DEADLEG LOSSES FROM A SIMULATED

DOMESTIC HOT WATER SYSTEM

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by

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T O   M Y   M O T H E R

ABSTRACT

This report considers the factors determining the deadleg losses of horizontal 'supply' hot water copper pipes, in a dynamic simulation rig of a domestic hot water system in the laboratory of the Mechanical Engineering Department at the University of Canterbury.

In order to simulate the calculated system daily deadleg losses for a real house for which the ambient temperature was low* and the room temperature was 18°C, an air-conditioning unit was used to supply cool air blowing through the five sections of 75 mm ID PVC air 'tunnels' which were built over the horizontal 'supply' hot water copper pipes [photo 1].

Thus the effect of deadleg losses due to natural-convection heat transfer in an ordinary domestic house was simulated experimentally by using forced-convection heat transfer in the rig.

Quantitative values of deadleg losses at different usage patterns and tank temperatures are tabulated.

*5.7°C as ambient temperature was being used in this project. It was the most severe daily average temperature of Christchurch in July (1960-1969) - data from Meteorological Office at the airport of Christchurch, New Zealand.

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CONTENTS

LIST OF SYMBOLS

		UNIT
A	Surface area	m^2
A_a	Cross-sectional area of the annulus	m^2
A_p	Pipe surface area	m^2
C	Thermal conductance	w/m^2K
C_p	Specific heat at constant pressure	$kJ/kg\ ^{o}C$
D	Diameter	m
D_E	Equivalent diameter	m
g	Acceleration of gravity	m/s^2
h	Heat transfer coefficient	w/m^2K
k	Thermal conductivity	w/mK
L	Pipe section length	m
$\dot{m}$	Mass rate of flow	kg/s
Mc	Mass per unit length of copper pipe	kg/m
$\dot{q}$	Rate of heat loss per meter	w/m
q	Heat loss per meter	kJ/m
q_D	Daily total loss per meter	kJ/m
q_{D_d}	Daily deadleg loss per meter	kJ/m
q_{D_f}	Daily loss per meter while water was flowing	kJ/m
$\dot{Q}$	Rate of heat loss	w
Q	Heat loss	kJ
Q_D	Daily total loss	kJ
Q_{D_d}	Daily deadleg loss	kJ
Q_{D_f}	Daily loss while water was flowing	kJ
Qd_T	System deadleg loss	kJ
Q_T	System total heat loss	kJ
ΣR	Total thermal resistance	K/W
t,T	Temperature	^{o}C, K

		UNIT
T	Time	min, sec
$[TC]$	Thermal capacity	J/K
T_f	Air film temperature	K
U	Overall heat transfer coefficient	w/K
V_w	Water volume per unit pipe length	m^3/m
Δx	Thickness, applied to building fabric	m
β	Volume coefficient of expansion	1/K
ε	Percentage difference	%
μ	Dynamic viscosity	kg/ms
ν	Kinematic viscosity, μ/ρ	m^2/s
ρ	Density	kg/m^3

Dimensionless Groups

Gr	Grashof No. $= \frac{g \beta (t_w - t_\infty) D^3}{\nu^2}$
Nu	Nusselt No. $= \frac{hD}{k}$
Pr	Prandtl No. $= \frac{C_p \mu}{k}$
Ra	Rayleigh No. $= Gr\, Pr$
Re	Reynolds No. $= \frac{UD}{\nu}$

CHAPTER 1

1. INTRODUCTION

Nationally, 40% of domestic electricity consumption is used in providing hot water. It is known that a large quantity of this energy is lost due to tank loss and deadleg loss in the system. The term "deadleg" means the length of pipe in which the water is stationary when there is no draw off.

The heat losses due to deadleg length has two sources

(a) Deadleg losses:
Heat loss in the deadleg length when the water is stationary.

(b) Flowing hot water heat losses:
Heat loss in the deadleg length when the water is flowing.

Flowing hot water heat losses is usually expected to be smaller than deadleg losses. When the water is stationary, deadleg losses lead to cooling of the water which results in a loss of energy and in more hot water being drawn off than is necessary, to "purge" the cooled water out of the pipe.

To experimentally determine these losses a series of experiments was undertaken on a domestic water system simulation rig [photo 2]. The existing rig is briefly described in Chapter 2. The typical schematic layout of the pipework arrangement is shown in Fig. 3.1. Modifications were made to this rig, described in detail in Chapter 4, to enable the following series of tests to be made.

A. Tank temperature at $70^{\circ}C$ under various water usage patterns with simulated deadleg cooling.

B. Tank temperature at $50^{\circ}C$ under various water usage patterns with simulated deadleg cooling.

C. Water quantity drawn off in different household devices of tank temperatures $70^{\circ}C$ and $50^{\circ}C$

CHAPTER 2

2. BRIEF DESCRIPTION OF THE RIG

Representative of a New Zealand domestic hot water system, it consisted of a hot water storage tank with electric immersion heater supplying the following user devices: basin, laundry/kitchen sink, washing machine, bath and shower. The rig also incorporated a waste water heat exchanger and a set of solar panels, but these were not used in the work reported here.

A SD System Z80 starter kit controls the hot water simulation rig [Photo 3]. Motortrol valves control the hot water, cold water inflows of water and drain valves at each device. Level sensors are used to detect changes in water levels of each device, while temperature changes are detected by LM334 temperature transducers (not existing in washing machine). Both these sensors are used for control with regard to the Z80.

Experiments were controlled by a micro-processor which contained programs to give 24-hour operation. The 24-hour data was recorded by a data logger which had 16 analog channels and 8 digital channels [Photo 4]. The data logger recorded temperature in terms of a voltage converted from a resistance input from platinum resistance thermometers. The voltage was converted to a temperature in the analysing program. 9 analog channels were used in this project. The 9 platinum resistance thermometer locations with their channel number are listed in Table 2.1. Water flow rates were measured by either dial readouts or by pulse generating Hall Effect Transistors. The number of pulses were recorded by the data logger digital channels. The digital channels of the data logger also recorded the time of a scan and the amount of electricity used (again in terms of pulses but from a light sensor on the kWh meter). The digital channels with the parameter they measured are listed in Table 2.2.

The data were analysed on the Department's HP2100A computer. The analysing program first counted the digital information to real parameters and then calculated the following energy quantities: energy in hot water from tank, energy in hot water at device, change in tank stored energy, electrical input. The tank loss was calculated by difference from:

Tank loss = Electrical input - energy in h.w. - change in stored energy

The program kept accumulative totals of all calculated quantities, and printed a summary of the information after 24 hours. In addition, the loss from the pipework system in 24 hours was estimated, again by difference, from:

System loss = Electrical input - energy in h.w. delivered - change in stored energy in tank - tank loss

To allow comparison between different tests all information was then normalized to a room temperature of 20^{o}C. In the following text, results referred to as 'computed' results are those from this analysis.

TABLE 2.1 The Location of the Platinum Resistance Thermometer

Analog Channel [#]	[Sign]	Location of the Platinum Resistance Thermometer (PRT)
1	PRT 1	Cold water supply to rig
7	PRT 7	Average cylinder hot water temperature
8	PRT 8	Cylinder hot water outlet
9	PRT 9	Hot water supply at basin
12	PRT 12	Hot water supply at laundry/kitchen sink
13	PRT 13	Hot water supply at washing machine
14	PRT 14	Hot water supply at bath
15	PRT 15	Hot water supply at shower
16	PRT 16	Air temperature

TABLE 2.2 The Parameter measured by Digital Channel

Digital Channel [#]	Parameter Measured
1	Time
2	Cold water supply to hot water cylinder
3	Cold water supply to system
4	Unused
5	Electricity used (kWh)

The data logger was set to scan its digital and analog channels every 10 minutes (master timer). If hot water was drawn off, then this triggered a slave timer which scanned every 20 seconds until hot water ceased to flow. Programs to analyse the 24-hour data were stored in the HP 2100A computer (Mechanical Engineering Department Computer).

In the existing rig, the hot water pipework was simply exposed to the laboratory atmosphere. The rig was modified so that losses from the pipework (deadleg losses) could be more correctly simulated. The pipework in real houses may pass through walls, under the floor or in roof spaces, it is exposed to much lower temperatures. The modifications to simulate this are described in Chapter 4.

CHAPTER 3

3. ESTIMATION OF DEADLEG LOSS FOR THE DOMESTIC HOT WATER PIPES

3.1 A Method to Estimate the Deadleg Loss for the 22 mm O.D. Domestic Hot Water Pipes at $70^{o}C$

Heat is transferred from one point to another whenever there exists a difference in temperature between the two points. The rate at which the heat is transferred depends upon the magnitude of the temperature difference, the distance between the two points and the type of material which separates them.

The calculation procedure of this part was first of all to calculate temperature of the unventilated air space condition assuming no hot water pipe was there, and then from the obtained temperature, the approximate heat loss from the hot water pipe was calculated.

(A) Consider the plane wall

As shown in Fig. 3.2a, a plane wall exposed to a warm fluid A on one side and a cooler fluid B on the other side. The heat transfer is expressed by

$$\dot{q} = h_1 A(t_A - t_1) = \frac{kA}{\Delta x}(t_1 - t_2) = h_2\, A\, (t_2 - t_B) \tag{3.1}$$

In order to simplify calculations it is assumed that the material is of even composition and that the rate of fall of temperature is constant for the thickness of the wall. The heat transfer process may be represented by the resistance network in Fig. 3.2b, and the overall heat transfer is calculated as the ratio of the overall temperature difference to the sum of the thermal resistances:

$$\dot{q} = \frac{t_A - t_B}{\frac{1}{h_1 A} + \frac{\Delta x}{kA} + \frac{1}{h_2 A}} \tag{3.2}$$

The value $\frac{1}{hA}$ is used to represent the convection resistance. The overall heat transfer by combined conduction and convection is frequently expressed in terms of an overall heat transfer coefficient U, defined by the relation

$$\dot{q} = U\, A\, \Delta t_{overall} \tag{3.3}$$

where the overall heat-transfer coefficient is

$$U = \frac{1}{\Sigma R} = \frac{1}{\frac{1}{h_1} + \frac{\Delta x}{k} + \frac{1}{h_2}} \tag{3.4}$$

(B) <u>Consider the water pipe</u>

The natural convection heat-transfer coefficients can be represented in the following functional form for a variety of circumstances:

$$Nu_f = C(Gr_f \; Pr_f)^m \tag{3.5}$$

where the subscript f indicates that the properties in the dimensionless groups are evaluated at the film temperature

$$T_f = \frac{T\infty + Tw}{2} \tag{3.6}$$

The product of the Grashof No. and Prandtl No. is called the Rayleigh No.

$$Ra = Gr_f \; Pr_f$$

where the Grashof No.

$$Gr_f = \frac{g \; \beta \; (t_w - t_\infty) D^3}{\nu^2} \tag{3.7}$$

The heat transfer coefficient may be evaluated from

$$h = \frac{k \; Nu_f}{D} \tag{3.8}$$

The volume coefficient of expansion β may be determined from tables of properties for the specific fluid. For ideal gases* it may be calculated from

$$\beta = \frac{1}{T} \tag{3.9}$$

where T is the absolute temperature of the gas.

* Air at atmospheric pressure may be treated as an ideal gas.

Example

An ordinary domestic house at Christchurch, New Zealand

An existing external wall constructed of 0.1 m concrete, 0.025 m air space, 0.1 m concrete, 0.1 m wooden frame and 0.0127 m plaster-board.

Use the undernoted data to calculate the deadleg loss for water at 70°C sitting in 22 mm O.D. copper pipe lying horizontally in the wooden frame as shown in Fig. 3.3. Room and ambient temperatures are 18°C and 5.7°C respectively.

Data

Wall surface area of unit pipe length

$$A = H \times 1 = 2.4 \text{ m}^2$$

Heat-transfer coefficient, inside air film [Ref. 1]

$$h_{ai} = 8 \frac{W}{m^2 K}$$

Thermal conductance, thickness 0.0127 m plastic-board [Ref. 2]

$$C = 12.78 \frac{W}{m^2 K}$$

Thermal conductivity, wooden frame [Ref. 3]

$$k_F = 0.28 \frac{W}{mK}$$

Pipe surface area of unit length

$$A_p = \pi D \times 1 = \pi \times 0.022 \text{ m}^2 = 0.069 \text{ m}^2$$

Thermal conductivity, concrete [Ref. 2]

$$k_c = 1 \frac{W}{mK}$$

Heat-transfer coefficient, 0.025 m air space [Ref. 1]

$$h_{as} = 2.8 \frac{W}{m^2 K}$$

Heat-transfer coefficient, outside air film [Ref. 4](for wind speed of 6.7 $\frac{m}{s}$)

$$h_{ao} = 34 \frac{W}{m^2 K}$$

Heat-transfer coefficient, 0.1 m non-ventilated air space [Ref. 2]

$$h_{as_{nv}} = 3.86 \frac{W}{m^2 K}$$

Area of wooden frame in wall 2.4 m x 1 m

$$A_F = 0.4 \text{ m}^2$$

Area of non-ventilated air space in wall 2.4 m x 1 m

$$A_{as_{nv}} = 2.0 \text{ m}^2$$

Answer

Air is a poor conductor of heat and the stagnant layer causes the surface temperatures to be higher than the general air temperatures. For air to air temperature difference, the profile is as shown in Fig. 3.4a. The air temperature in the wooden frame is considered as the average temperature of t_1 and t_2 which are shown in the resistance analogy network in Fig. 3.4b.

From equation 3.3, the flow of heat is

$$\dot{q} = UA(t_{ai}-t_{ao})$$

From equation 3.4, the overall heat-transfer coefficient is

$$U = \frac{1}{\Sigma R}$$

where the total thermal resistance

$$\Sigma R = \frac{1}{h_{ai}A} + \frac{1}{CA} + \frac{1}{h_{as_{av}}A_{as_{av}} + \frac{k_F A_F}{\Delta X_F}} + \frac{\Delta X_c}{k_c A} + \frac{1}{h_{as}A} + \frac{\Delta X_c}{k_c A} + \frac{1}{h_{ao}A}$$

$$= \frac{1}{8A} + \frac{1}{12.78A} + \frac{1}{3.86 \times 2 + \frac{0.28A_F}{0.1}} + \frac{0.1}{A} + \frac{1}{2.8A} + \frac{0.1}{A} + \frac{1}{34A}$$

$$= \left(\frac{1}{8} + \frac{1}{12.78} + \frac{0.1}{1} + \frac{1}{2.8} + \frac{0.1}{1} + \frac{1}{34}\right)\frac{1}{2.4} + \frac{1}{3.86 \times 2 + \frac{0.28}{0.1} \times 0.4}$$

$$= 0.44 \qquad \left[\frac{K}{W}\right] \qquad (3.10)$$

By voltage divider analogy,

$$\frac{18-t_1}{\frac{1}{8A} + \frac{1}{12.78A}} = \frac{18 - 5.7}{\Sigma R}$$

$$\frac{18-t_1}{0.085} = \frac{12.3}{\Sigma R}$$

$$18-t_1 = \frac{1}{\Sigma R}$$

$$t_1 = 18 - \frac{1}{\Sigma R} \qquad (3.11)$$

$$\frac{t_2 - 5.7}{\left(\frac{0.2}{1} + \frac{1}{2.8} + \frac{1}{34}\right) \frac{1}{2.4}} = \frac{18-5.7}{\Sigma R}$$

$$t_2 - 5.7 = \frac{3}{\Sigma R}$$

$$t_2 = 5.7 + \frac{3}{\Sigma R} \qquad (3.12)$$

* Insert from page 9 and 10

The air temperature in the wooden frame t_∞ is 14°C. From equation 3.6, the film temperature

$$t_f = \frac{70^\circ C + 14^\circ C}{2} = 42^\circ C$$

$$T_f = (273 + 42)K = 315 \text{ K}$$

From physical tables, air at 315 K

$$\beta = \frac{1}{T_f} = \frac{1}{315} = 3.175 \times 10^{-3} \text{ K}^{-1}$$

$$Pr_f = 0.7034$$

$$\nu = 1.7114 \times 10^{-5} \frac{m^2}{s}$$

$$k = 2.739 \times 10^{-2} \frac{w}{mK}$$

The Grashof No. Gr_f from equation 3.7 is

$$Gr_f = \frac{g \beta (t_w - t_\infty) D^3}{\nu^2}$$

$$= \frac{9.81 \times 3.175 \times 10^{-3} (70-14) 0.022^3}{(1.7114 \times 10^{-5})^2}$$

$$= 63411$$

So $Gr_f Pr_f = 63411 \times 0.7034 = 44603$

Since $10^4 < 44603 < 10^9$

Therefore $C = 0.53 \quad m = \frac{1}{4}$ [Ref. 3]

From equation 3.5, the Nusselt No. Nu_f is

$$Nu_f = C(Gr_f \ Pr_f)^m$$

$$= 0.53 \ (44603)^{\frac{1}{4}}$$

$$= 7.702$$

From equation 3.8, the heat transfer coefficient h is

$$h = \frac{kNu_f}{D}$$

$$= \frac{2.739 \times 10^{-2}}{0.022} \times 7.702 = 9.59 \ \frac{w}{m^2 K}$$

* and from equation 3.11, 3.12

$$t_1 = 18 - \frac{1}{\Sigma R}$$

$$= 18 - \frac{1}{0.44}$$

$$= 15.73 \ ^{\circ}C$$

$$t_2 = 5.7 + \frac{3}{\Sigma R}$$

$$= 5.7 + \frac{3}{0.44}$$

$$= 12.52 \ ^{\circ}C$$

Therefore, the air temperature of the wooden frame is

$$t_\infty = \frac{t_1 + t_2}{2}$$

$$= \frac{15.73 + 12.52}{2}$$

$$= 14^\circ C$$

Hence, from equation 3.1, the rate of deadleg loss for water at 70°C is

$$\dot{q} = h \pi D(t_w - t_\infty)$$

$$= 9.59 \pi (0.022)(70-14)$$

$$= 37 \frac{W}{m}$$

3.2 Analysis for Predicting Deadleg Loss for Various Water Temperatures and Pipe Sizes

The schematic layout of the pipework arrangement of the dynamic simulation of a domestic hot water system on the rig is shown in Fig. 3.1. The deadleg lengths of all the sections are shown in Table 3.1.

TABLE 3.1 Deadleg Lengths on the Experimental Rig

Section	Deadleg Length [m]
TK-J1	(3.43)
0-J1	1.5
J1-J2	2.1
J2-J3	1.35
J3-J4	1.2
J4-J5	3

(A) Modification to the first deadleg length

Only the main horizontal transmission lengths on the rig were used for the low ambient temperature deadleg simulation. For section TK-J1, the length from TK to 0 was well insulated with "Insultube" rigid urethane foam sleeving to make its heat loss as small as possible, and the length 0-J1 was modified to give correct water quantity and heat loss.

$$\frac{\pi}{4} D_i^2 \times 1.5 = \frac{\pi}{4} (0.020)^2 \times 3.43$$

$$D_i = 0.030 \text{ m}$$

This modification was done to maintain a simple horizontal air tunnel.

(B) Calculation of rate of deadleg loss as a function of water temperature

(i) For 22 mm O.D. copper pipes

In Table 3.2 is listed the physical properties of air for water temperatures from 70°C to 14°C. As illustrated in Chapter 3.1, the rate of water deadleg losses corresponding to the fall of water temperatures can be calculated and the results are listed in the second column of Table 3.3. The results are also plotted in Fig. 3.5.

(ii) For 32 mm O.D. copper pipe

As a result of the modification of the first deadleg length in (A) above, the required rate of deadleg losses in that particular length is obtained by multiplying the ratio of the original and modified deadleg length to the heat losses of the 22 mm O.D. copper pipe, i.e.

$$\left[\dot{q}\right]_{32 \text{ mm}} = \frac{3.430}{1.5} \left[\dot{q}\right]_{22 \text{ mm}} \qquad (3.13)$$

The results are listed in the third column of Table 3.3. For convenience, the rate of deadleg losses versus water temperature with various pipe sizes is plotted in Fig. 3.5.

TABLE 3.2 Physical Properties of Air at Various Film Temperatures T_f Ref. [5]

t_s $[^\circ C]$	t_∞ $[^\circ C]$	$t_f = \frac{t_s + t_\infty}{2}$ $[^\circ C]$	T_f $[K]$	$\beta = \frac{1}{T_f}$ $10^{-3} \times [K^{-1}]$	Pr	ν $10^{-5} \times [\frac{m^2}{s}]$	k $10^{-2} \times [\frac{W}{mK}]$
70	14	42	315	3.175	0.7034	1.711	2.739
60	14	37	310	3.226	0.7046	1.664	2.701
50	14	32	305	3.279	0.7058	1.616	2.662
40	14	27	300	3.333	0.707	1.568	2.624
30	14	22	295	3.390	0.7082	1.523	2.585
20	14	17	290	3.448	0.7094	1.478	2.546
15	14	14.5	287.5	3.478	0.71	1.456	2.526

TABLE 3.3 Deadleg Loss with Various Water Temperatures and Pipe Sizes

Water Temp. t $[^\circ C]$	Deadleg Loss $\dot{q}$ 22 mm O.D. $[\frac{W}{m}]$	Deadleg Loss $\dot{q}$ 32 mm O.D. $[\frac{W}{m}]$
70	37	84.6
60	29	66.3
50	21.6	49.4
40	14.4	32.9
30	7.9	18.1
20	2.4	5.5
15	0.27	0.6
14	0	0

(C) Calculation of cooling time $[\Delta T]$ for every 10 deg. C water temperature fall $[\Delta t]$

Using the above data on the rate of heat loss varying with temperature, two cooling curves can now be calculated by means of an energy balance between the water and the surrounding air. The energy equation is

$$\begin{bmatrix}\text{Average}\\ \text{Thermal}\\ \text{Capacity}\\ \text{of Water}\\ \text{and Copper}\\ \text{Pipes}\end{bmatrix} \begin{bmatrix}\text{Water}\\ \text{Temperature}\\ \text{Fall}\end{bmatrix} = \begin{bmatrix}\text{Average}\\ \text{Rate of}\\ \text{Deadleg}\\ \text{Heat}\\ \text{Loss}\end{bmatrix} \begin{bmatrix}\text{Cooling}\\ \text{Time}\\ \text{Interval}\end{bmatrix}$$

$$[TC]\ \Delta t = [\dot{q}]\ \Delta T$$

$$[c_{p_w} V_w \rho_w + c_{p_c} M_c]\ \Delta t = [\dot{q}]\ \Delta T \qquad (3.14)$$

(i) For 30 mm I.D. copper pipe

Data

$$C_{p_w} = 4.18 \frac{kJ}{kg\ K}$$

$$V_w = \frac{\pi D^2}{4} \times 1 = \frac{\pi (0.03)^2}{4} = 7.07 \times 10^{-4} \frac{m^3}{m}$$

$$C_{p_c} = 0.3925 \frac{kJ}{kg\ K}$$

$$M_c = 0.91 \frac{kg}{m}$$

From energy equation 3.14 and putting $\Delta t = 10$ deg. C

$$\left[C_{p_w} V_w \rho_w + C_{p_c} M_c\right] \Delta t = \left[\bar{\dot{q}}\right] \Delta T$$

$$\left[4.18 \times 7.07 \times 10^{-4} \times \rho_w + 0.3925 \times 0.91\right] 10 = \left[\bar{\dot{q}}\right] \Delta T$$

$$\Delta T = \frac{3.6 + 2.96 \times 10^{-2} \rho_w}{\bar{\dot{q}}} \tag{3.15}$$

(ii) For 20 mm I.D. copper pipes

Data

$$C_{p_w} = 4.18 \frac{kJ}{kg\ K}$$

$$V_w = \frac{\pi D^2}{4} \times 1 = \frac{\pi (0.02)^2}{4} = 3.14 \times 10^{-4} \frac{m^3}{m}$$

$$C_{p_c} = 0.3925 \frac{kJ}{kg\ K}$$

$$M_c = 0.71 \frac{kg}{m}$$

From energy equation 3.14 and putting $\Delta t = 10$ deg. C.

$$\left[C_{p_w} V_w \rho_w + C_{p_c} M_c\right] \Delta t = \left[\bar{\dot{q}}\right] \Delta T$$

$$\left[4.18 \times 3.14 \times 10^{-4} \times \rho_w + 0.3925 \times 0.71\right] 10 = \left[\bar{\dot{q}}\right] \Delta T$$

$$\Delta T = \frac{2.79 + 1.3 \times 10^{-2} \rho_w}{\bar{\dot{q}}} \tag{3.16}$$

In order to calculate the cooling time interval for every 10 deg. C. water temperatures fall from 70^{o}C to 14^{o}C, firstly, read off the average deadleg heat loss $\left[\bar{\dot{q}}\right]$ at the average water temperature $\left[\bar{t}\right]$ of every temperature interval in Fig. 3.5. Then from equations 3.15, 3.16, the cooling time interval can be calculated. These results have been listed in Table 3.4. The values of water density $\left[\rho_w\right]$ in Table 3.4 correspond to the average water temperature. The cooling curves for the two pipe sizes are plotted in Fig. 3.6, for an initial water temperature of 70^{o}C.

TABLE 3.4 Cooling Time for Every 10 Deg. C. Water Temperature Drop (Except the Last Row)

Water Temperature			Water Density	I.D. = 0.03 m		I.D. = 0.02 m	
				Rate of Deadleg Loss	Cooling Time	Rate of Deadleg Loss	Cooling Time
t_1 [°C]	t_2 [°C]	$t = \frac{t_1+t_2}{2}$ [°C]	$\rho_w = \frac{1}{v_f}$ [kg/m^3]	$\dot{q}$ [$\frac{W}{m}$]	ΔT [min]	$\dot{q}$ [$\frac{W}{m}$]	ΔT [min]
70	60	65	980	75.5	7.2	33	7.8
60	50	55	986	58	9.4	25	10.4
50	40	45	990	41	13.4	17.5	14.9
40	30	35	994	25	22	11	23.8
30	20	25	997	11	50.2	5	52.5
20	14	17	999	2.5	132.7	1.2	131.5

CHAPTER 4

4. DESCRIPTION OF THE MODIFICATIONS TO THE RIG

4.1 Materials Used and Description of the Modifications

(A) Non-experimental section of pipework

Section TK-0 was covered by 'Insultube' rigid urethane foam sleeving for minimising the heat loss from this section.

(B) 20 mm I.D. and 30 mm I.D. copper pipes

The 'Qest-Dux' polybutylene pipes were changed to copper pipes so that surface temperature measurement would be a good indication of the water temperature. Heat transfer could also be more easily controlled.

(C) Thermocouples along the horizontal 'supply' hot water pipes

3 copper-constantan thermocouples were attached to the outer surface of the copper tube at the inlet, middle and outlet of each of the 5 sections of pipe. A thermocouple was also located at the storage tank outlet (see Fig. 3.1). The thermocouples were connected to an ice-junction and a digital millivolt meter through a 16-channel manually controlled switch.

All the exposed junction points between the thermocouples and copper pipes were covered with a thermal insulant to avoid temperature fall.

(D) 75 mm I.D. PVC 'tunnels'

5 sections of air 'tunnels' over the copper pipes through which air could be blown were installed as sketched in Fig. 3.1. Every tunnel was supported centrally about the copper pipes by two perspex discs used as end caps for that section.

(E) Air flow control

4 holes around the end part of every PVC 'tunnel' were used to control the air flow rate by sliding a collar, which was made of clear acetate sheet, to partly cover the holes.

(F) 64 mm I.D. 'dunlopflex' ducting hose

5 flexible hoses transmitted cool air from the air-conditioning unit to the 5 sections of PVC air 'tunnels'. They were connected to the tunnels at the inlet ends of the hot water supply, giving 5 independent parallel flow heat-exchangers (see Fig. 4.1 and 3.1).

(G) Air-conditioning unit

The mean perimeter of the circulated air ducting was 3.6 m and the cross-sectional area of the ducting was 0.46 x 0.28 m^2. The air-conditioning unit could provide cool air at about 10°C. The supply air temperature can be changed by adjusting the pressure of the expansion valve of the refrigeration system (see photo 1).

4.2 Determining the Size of the PVC Air Tunnels

In order to simulate the deadleg losses through the building fabric, PVC tunnels were built over the horizontal pipes through which air could be blown. The heat-transfer coefficient for the hot water pipe was determined by equation 3.8. For flow in an annulus it is customary to define an 'equivalent diameter' as four times the cross-sectional flow area divided by the wetted perimeter. The equivalent diameter is the diameter of a hypothetical circular pipe which has the same pressure gradient along its axis as the noncircular ducts if the mean flow velocities are the same. If the annulus has an inner diameter (i.e., the O.D. of the inner pipe) of D_1, and an outer diameter (i.e., the I.D. of the outer pipe) of D_2, then the equivalent diameter, D_E, is

$$D_E = 4\left[\frac{\frac{\pi}{4}\,(D_2^2-D_1^2)}{\pi\,(D_2+D_1)}\right] = D_2-D_1 \tag{4.1}$$

For forced-convection inside an annular space the Nusselt No. and Reynolds No. are based on this equivalent diameter.

$$Nu_{D_E} = \frac{h\,D_E}{k} \tag{4.2}$$

$$Re_{D_E} = \frac{U_m\,D_E}{\nu} \tag{4.3}$$

The heat-transfer coefficient of the annulus can be evaluated by Nusselt No. based on the equivalent diameter.

$$Nu_{D_E} = 0.023(Re_{D_E})^{0.8}\ Pr^n \tag{4.4}$$

$$n = 0.4, \text{ heating.}$$

[Ref. 6]

Concentric-tube, parallel flow heat exchanger

This section describes a simple approach which was used to estimate the heat exchanger size to accomplish the heat loss requirement as tabulated in Table 3.3. The calculations were based on a forced-convection heat transfer.

On the experimental rig, the 'Qest-Dux' polybutylene pipes were replaced by copper pipes so that surface temperature measurement would be a good indication of the water temperature. Heat transfer could also be more easily controlled. A simple concentric tube parallel flow heat exchanger system was determined to apply while the water was sitting in copper pipes during the running of the experiments for water deadleg heat losses. A schematic layout of the heat exchanger was shown in Fig. 4.1.

From the standpoint of the deadleg losses required for water temperature of 70^{o}C,in the annular space between the air tunnels and water pipes, as estimated in Chapter 3.2(B), a very rough requirement of the mass flow rate of cooling air in both sizes of copper pipes was estimated.

Energy balance required

$$\dot{q} = \dot{m}\ C_p(t_{c_o} - t_{c_i}) \tag{4.5}$$

Therefore; the mass flow rate required

$$\dot{m} = \frac{\dot{q}}{C_p(t_{c_o} - t_{c_i})} \tag{4.6}$$

From equation 4.1, the equivalent diameter is

$$D_E = D_i - d_o \tag{4.7}$$

where D_i = the inside diameter of the air tunnels

d_o = the outside diameter of the copper pipes.

The cross-sectional area of the annulus

$$A_a = \frac{\pi}{4} (D_i^2 - d_o^2) \tag{4.8}$$

The mean flow velocity

$$U_m = \frac{\dot{m}}{\rho A} \tag{4.9}$$

A tabulation of physical properties of air at various temperatures which will be encountered in this chapter is noted in Table 4.1.

TABLE 4.1 Air Properties at Low Pressure [Ref. 5]

Air Temperature		Density	Kinematic Viscosity	Prandtl No.	Thermal Conductivity
t_a [^oC]	t_a [K]	ρ $\left[\frac{kg}{m^3}\right]$	ν $\times 10^{-5} \left[\frac{m^2}{s}\right]$	Pr -	k $\times 10^{-2} \left[\frac{W}{mK}\right]$
13	286	1.237	1.778	0.7104	2.514
14	287	1.233	1.783	0.710	2.522
15	288	1.228	1.788	0.7099	2.53
15.5	288.5	1.226	1.790	0.7098	2.534
16.5	289.5	1.222	1.795	0.7095	2.542

To estimate the size of the outer pipe of the air tunnel, it was assumed that the average air temperature within the tunnel would be the same as the air-space in the house wall calculations in Chapter 3, i.e. 14^{o}C.

Thus, for an inlet air temperature of 10^{o}C, the outlet temperature was assumed to be 18^{o}C.

The heat losses from the copper pipes were taken from Table 3.3.

Suitable sizes of commercially available PVC pipes were 0.060 m, 0.075 m, 0.095 m inside diameter and of these, the 0.075 m diameter pipe was selected. The larger diameter pipe would give flow not in

TABLE 4.2 The Predicting Value of Air Characteristics by Using I.D. = 0.075 m PVC Air Tunnels

Copper Pipe Outside Diameter	Air Mass Flow Rate	Equivalent Diameter	Annulus Cross-Sectional Area	Mean Air Flow Velocity	Reynolds No.	Nusselt No.	Heat Transfer Coefficient
d_o $[m]$	$\dot{m}$ $\left[\frac{kg}{sec}\right]$	D_E $[m]$	A_a $[m^2]$	U_m $\left[\frac{m}{sec}\right]$	Re_{D_E}	Nu_{D_E}	h $\left[\frac{w}{m^2 K}\right]$
0.022	4.61×10^{-3}	0.053	4.04×10^{-3}	0.925	3379	13.3	6.3
0.032	1.05×10^{-2}	0.043	3.61×10^{-3}	2.366	7013	23.9	14

the turbulent regime, and the smaller pipe would lead to too large a pressure drop.

4.3 Calculation of Required Air Flow Rates

In order to find the air mass flow rate which produces the required heat loss from the copper pipes (see Table 3.3), the following equations have to be satisfied simultaneously.

Neglecting heat loss from the outside of the PVC pipe, then :

Heat gain by air $\dot{Q} = \dot{m}\, C_p\, (t_{a_o} - t_{a_i})$ (4.10)

Heat loss by pipe $\dot{Q} = h\, A_s\, (t_w - \bar{t}_a)$ (4.11)

where $\bar{t}_a = \dfrac{t_{a_i} + t_{a_o}}{2}$ (4.12)

Forced convection heat-transfer coefficient

$$h = \frac{k\, Nu_{D_E}}{D_E} \quad (4.13)$$

$$\text{and} \quad Nu_{D_E} = 0.023\, (Re_{D_E})^{0.8}\, Pr^{0.4} \quad (4.14)$$

$$\text{Reynolds No.} \quad Re_{D_E} = \frac{\rho\, U_m\, D_E}{\mu} = \frac{D_E}{\mu}\,\frac{\dot{m}}{A_a} \quad (4.15)$$

Since the air-conditioning unit delivered air at $10^{o}C$, t_{a_i} was set to this value. The value of $\dot{Q}$ was known for $t_w = 70^{o}C$ from Table 3.3. The calculation technique was therefore to assume a value for $\bar{t}_a$ and to use equation 4.11 to 4.15 to calculate $\dot{m}$. This was then used in equation 4.10 to compute t_{a_o} and hence $\bar{t}_a$, which was then compared with the originally assumed value. The process was iterated until consistent values were obtained.

(A) Pipe section 0-J1

e.g. $D_i = 0.075$ m $\quad d_o = 0.032$ m

$D_E = D_i - d_o = 0.043$ m

$A_a = \frac{\pi}{4}(D_i^2 - d_o^2) = 3.61 \times 10^{-3}\ m^2$

Assume $\bar{t}_a = 15.5^{\circ}C$, from equation 4.11, the heat transfer coefficient

$$h = \frac{\dot{Q}}{A_s(t_w - \bar{t}_a)}$$

$$= \frac{84.6 \times 1.5}{\pi \times 0.032 \times 1.5 \times (70-15.5)}$$

$$= 15.4 \ \frac{W}{m^2 K}$$

From equation 4.13, the Nusselt No.

$$Nu_{D_E} = \frac{h \ D_E}{k}$$

$$= \frac{15.4 \times 0.043}{2.534 \times 10^{-2}}$$

$$= 26.2$$

From equation 4.14, the Reynolds No.

$$Re_{D_E} = \log^{-1} \left[\frac{1}{0.8} \log \frac{Nu_{D_E}}{0.023 \ Pr^{0.4}} \right]$$

$$= \log^{-1} \left[\frac{1}{0.8} \log \frac{26.2}{0.023 \times 0.7098^{0.4}} \right]$$

$$= 7855$$

From equation 4.15, the air mass flow rate

$$\dot{m} = \frac{Re_{D_E} \ A_a \ \mu}{D_E}$$

$$= \frac{7855 \times 3.61 \times 10^{-3} \times 1.790 \times 10^{-5}}{0.043}$$

$$= 1.18 \times 10^{-2} \ \frac{kg}{s}$$

From equation 4.10, the outlet temperature of the cooling air

$$t_{a_o} = \frac{\dot{Q}}{\dot{m}C_p} + t_{a_i}$$

$$= \frac{84.6 \times 1.5}{1.18 \times 10^{-2} \times 1005} + 10$$

$$= 20.7^{\circ}C.$$

From equation 4.12, the average temperature of the cooling air

$$\bar{t}_a = \frac{t_{a_i} + t_{a_o}}{2}$$

$$= \frac{10 + 20.7}{2}$$

$$= 15.4^{\circ}C$$ (only $0.1^{\circ}C$ less than the assumption).

(B) Remaining pipe sections

e.g. $D_i = 0.075$ m $\quad d_o = 0.022$ m

$D_E = D_i - d_o = 0.053$ m

$A_a = \frac{\pi}{4}(D_i^2 - d_o^2) = 4.04 \times 10^{-3}\ m^2$

For pipe section J1-J2, assume $\bar{t}_a = 15^{\circ}C$, the heat transfer coefficient

$$h = \frac{\dot{Q}}{A_s(t_w - \bar{t}_a)}$$

$$= \frac{37 \times 2.1}{\pi \times 0.022 \times 2.1 \times (70-15)}$$

$$= 9.7\ \frac{w}{m^2K}$$

Nusselt No.

$$Nu_{D_E} = \frac{h\ D_E}{k}$$

$$= \frac{9.7 \times 0.053}{2.53 \times 10^{-2}}$$

$$= 20.3$$

Reynolds No.

$$Re_{D_E} = \log^{-1}\left[\frac{1}{0.8}\log\frac{Nu_{D_E}}{0.023\ Pr^{0.4}}\right]$$

$$= \log^{-1}\left[\frac{1}{0.8}\log\frac{20.3}{0.023 \times 0.7099^{0.4}}\right]$$

$$= 5709.7$$

The air mass flow rate

$$\dot{m} = \frac{Re_{D_E}\ A_a\ \mu}{D_E}$$

$$= \frac{5709.7 \times 4.04 \times 10^{-3} \times 1.788 \times 10^{-5}}{0.053}$$

$$= 7.8 \times 10^{-3}\ \frac{kg}{s}$$

The outlet temperature of the cooling air

$$t_{a_o} = \frac{\dot{Q}}{\dot{m}\ C_p} + t_{a_i}$$

$$= \frac{37 \times 2.1}{7.8 \times 10^{-3} \times 1005} + 10$$

$$= 20^{o}C$$

The average temperature of the cooling air

$$\bar{t}_a = \frac{t_{a_i} + t_{a_o}}{2}$$

$$= \frac{10 + 20}{2}$$

$$= 15^{o}C \qquad \text{(same as the assumption)}$$

Repeating the same procedures as above, the outlet and average temperatures of the cooling air, the heat-transfer coefficient and the air mass flow rate of the remaining pipe sections were obtained. The results are listed in Table 4.3. These parameters were significant for determining the requirements of the air-conditioning unit from which cool air was supplied to cool the hot water pipe.

TABLE 4.3 <u>Air Required Condition With I.D. = 0.075 m Air Tunnel</u>

Section	Heat Loss	Pipe Length	Air Outlet Temperature	Air Average Temperature	Heat Transfer Coefficient	Air Mass Flow Rate
	$\dot{q}$ $\left[\frac{W}{m}\right]$	L $[m]$	t_{C_o} $[{}^oC]$	$\bar{t}_a$ $[{}^oC]$	h $\left[\frac{W}{m^2K}\right]$	$\dot{m}$ $\left[\frac{kg}{s}\right]$
0-J1	84.6	1.5	20.7	15.4	15.4	1.19×10^{-2}
J1-J2	37	2.1	20	15	9.7	7.8×10^{-3}
J2-J3	37	1.35	16.5	13.3	9.5	7.6×10^{-3}
J3-J4	37	1.2	16	13	9.4	7.5×10^{-3}
J4-J5	37	3	23.5	16.8	10	8.1×10^{-3}

4.4 The Ducting Connecting the Air Tunnels to the Air Conditioner

For convenience, this ducting had to be flexible, easily connected both to the air-conditioner duct and to the air tunnels, and not impose large pressure drops.

'Dunlopflex' of 0.064 m I.D. was selected as suitable. This size allowed the connections to the air-conditioner duct to be in-line across the supply opening, giving reasonably uniform air flow.

In preliminary tests, it was found that the air-conditioner could produce 3 times the flow rates required as in Table 4.3, so pressure drop in these hoses was not a problem.

4.5 Air Flow Rate Control in the Air Tunnels

The flow rate was controlled by sliding a collar over discharge holes at the end of each tunnel. The holes were sized to give approximately the same area as the annular cross-sections. Four equi-spaced 0.04 m holes were used, and these were found to be satisfactory for the tests.

CHAPTER 5

5. EXPERIMENTAL WORK

5.1 Final Setting of the Cooled Air Flow Rate

In the previous chapter, the air tunnels and flow conditions were sized on the basis of equivalence between natural convection in the real location and forced convection in the rig for water in the pipework at 70°C.

However, as the water cools, the rates of heat loss will change because of the different natures of natural and forced convection and in fact the water in the rig would be cooled too rapidly.

It was therefore necessary to "fine-tune" the apparatus so that on average the total heat loss over a given period would equal the calculated value. This chapter describes the methods employed.

(A) General procedure

The 'fine-tuning' was done with the rig operating on an average water usage pattern (see Table A(1)) and with storage tank thermostat settings of 70°C and 50°C. The middle thermocouple of each pipe section was recorded over a 24-hour period, and the results are shown in Table C(1), C(2) and the results for pipe section O-J1 are plotted in Fig. 5.1 for 70°C tank temperature.

From these, it can be seen that each set of results can be divided into two parts (1) the periods during which water was flowing, and hence the pipe temperature was approximately constant, and (2) the remaining periods when the pipe (and water) was cooling.

Variation of rates of heat loss with temperature for the natural convection conditions have already been computed in Chapter 3.2 (Fig. 3.5). These results were used to compute what the natural convection heat loss would be for each pipe section over a 24-hour period.

The actual results (Fig. 5.1) were then used to estimate the forced convection heat transfer coefficients so that heat loss during the constant temperature periods could be calculated. The actual heat loss from the pipes and water during the cooling periods was also calculated from the experimental cooling curves.

The air flow rate to each pipe section was adjusted until the total heat loss in forced convection equalled that calculated to occur under natural convection.

(B) Determination of heat-transfer coefficient values from experimental cooling curves

For a section of a hot water pipe, energy balance over a short time dT at any time T from the commencement of cooling is

$$\begin{bmatrix}\text{Energy lost by}\\ \text{water and pipe}\end{bmatrix} = \begin{bmatrix}\text{Energy transferred to air}\\ \text{by forced convection}\end{bmatrix}$$

so $$[TC]\ \Delta t_w = -h\ A_p (t_w - t_a)\ dT$$

Rearranging and integrating gives

$$\int_o^T -\frac{h\ A_p}{[TC]}\ dT = \int_{t_w \text{ at } T=0}^{t_w \text{ at } T=T} \frac{dt_w}{t_w - t_a}$$

If h, $[TC]$ and t_a are assumed constant then

$$-\frac{h\ A_p}{[TC]}\ T = \Big[\ln(t_w - t_a)\Big]_{t_w \text{ at } T=0}^{t_w \text{ at } T=T}$$

If $t_w = t_{w_o}$ at T = 0 then

$$-\frac{h\ A_p}{[TC]}\ T = \ln\left[\frac{t_w - t_a}{t_{w_o} - t_a}\right]$$

Thus $$h = \frac{[TC]}{A_p\ T}\ \ln\left[\frac{t_{w_o} - t_a}{t_w - t_a}\right] \qquad (5.1)$$

Actual heat loss quantities were computed from the values of h determined from equation 5.1 using the following relation

$$Q = \sum_{T=0}^{T=T} (h\ A_p (t_w - t_a))\ \Delta T$$

Since the variation of h with t_w (and hence time) was being investigated this was the method used to obtain Q. It could have been obtained more directly from $Q = [TC]\ \Delta t_w$ had the calculations for h not been needed.

5.2 Calculated Heat Loss With Natural Convection Cooling; Tank Temperature = 70°C Nominal

(A) Pipe section 0-J1

Using the cooling curve results from Chapter 3, with an initial temperature of 70°C, the following table giving the time interval and total heat loss was constructed.

TABLE 5.1a

Time	Mean Water Temperature in Time Interval	Mean Rate of Heat Loss Over Time Interval	Heat Loss in Time Interval
T [min]	t_w [°C]	$\dot{q}$ [$\frac{W}{m}$]	q_d [$\frac{kJ}{m}$]
0			
	53	54.5	65.4
20			
	38	30	36
40			
	29.5	17	20.4
60			
	24	10	12
80			
	21	6.5	7.8
100			
	18.5	3.7	4.4
120			
	17	2.5	3.0
140			
	16	1.7	2
160			
	14.6	0.5	0.6
180			
	14.1	0.1	0.12
200			
	14.05	0.05	0.06
220			
		Total q_d =	151.8

This relation between heat loss and time was then applied to all the cooling parts of the experimental data with an average water usage pattern (see Appendix A, Table A(1)) to obtain the natural convection heat loss. For example, for a cooling time of 9.5 minutes, the heat loss is $\frac{9.5}{20}$ x 65.4. The results are listed in Table 5.1b.

TABLE 5.1b

User Device Operated	Cooling-Time Interval	Heat Loss From Pipe and Water
	T [min]	$\left[\frac{kJ}{m}\right]$
Shower	9.5	31
Basin	14.5	47.4
Shower	24.5	73.5
L/Sink	13	42.5
K/Sink	13	42.5
Basin	44.5	106
Washing M/c	238.5	151.8
K/Sink	358	151.8
K/Sink	13	42.5
Basin	104.5	142.6
Bath	582	151.8
	Total q_{D_d}	= 983.4

To the above loss, there must be added the heat lost while water was flowing, to obtain the total for the day.

From Fig. 3.5, the calculated rate of heat loss when $t_w = 65.1^oC$ was $75.5\ \frac{W}{m}$. The total time for which water was flowing was 25 minutes (see Appendix A, Table A(1)).

Hence, heat loss during this period $q_{D_f} = 75.5 \times 25 \times 60 \times 10^{-3}$

$$= 113.3\ \frac{kJ}{m}$$

Thus, the daily total heat loss, as calculated for natural convection cooling, was

$$q_D = q_{D_d} + q_{D_f}$$

$$= 983.4 + 113.3$$

$$= 1096.7\ \frac{kJ}{m}$$

(B) Remaining pipe sections

Using the same technique as in (A) above, but with the appropriate initial water temperature (see Table C(1)) and cooling intervals, the heat loss for each of the other sections of pipe was calculated, and the results are summarised below.

TABLE 5.2

Pipe Section	Daily Heat Loss in Natural Convection		
	Deadleg Losses	While Water was Flowing	Total
	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_{D_f} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$
J1-J2	440.3	47	487.3
J2-J3	252.5	31.3	283.8
J3-J4	184.3	26.5	210.8
J4-J5	88.5	17.3	105.8

Total for all sections 0 to J5 = 2184.4

5.3 Calculated Heat Loss With Natural Convection Cooling; Tank Temperature = 50°C Nominal

The calculations of Chapter 5.2 were repeated for a tank temperature of 50°C. The results are summarised below.

TABLE 5.3

Pipe Section	Daily Heat Loss in Natural Convection		
	Deadleg Losses	While Water was Flowing	Total
	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_{D_f} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$
0-J1	612.6	66.8	679.4
J1-J2	271	28.2	299.2
J2-J3	157.8	18.6	176.4
J3-J4	115	16	131
J4-J5	53.1	10.2	63.3

System total = 1349.3

5.4 Calculation of Actual Heat Loss With Forced Convection Cooling; Tank Temperature = $70^{\circ}C$ Nominal

(A) Pipe section 0-J1

From the experimental cooling curves, and using the method outlined in Chapter 5.1(B), the forced convection heat transfer coefficient was calculated at different stages in the cooling process. For the calculations $[TC] = 3300\ L\ \frac{J}{K}$ (see Appendix B), $A_p = 0.1005\ L\ m^2$ were used for the 0.03 diameter pipe.

TABLE 5.4a

Time	Forced Convection Heat-Transfer Coefficient	Heat Loss in Time Interval
T [min]	h $\left[\frac{W}{m^2K}\right]$	q_d $\left[\frac{kJ}{m}\right]$
0		
20	17.6	68.8
40	16.3	36
60	16.5	19.7
80	15.7	11.4
100	16.2	6.1
120	14.8	4.1
140	14.9	2.3
160	15.7	1.1
		Total q_d = 149.5

It should be noted that the heat-transfer coefficient is very nearly of constant value, the mean value being $16\ \frac{W}{m^2K}$.

Applying the relation between time and heat loss to the actual cooling curves throughout the day's operations with an average water usage pattern gave the following results.

TABLE 5.4b

User Device Operated	Cooling-Time Interval	Heat Loss from Pipe and Water
	T [min]	$\left[\frac{kJ}{m}\right]$
Shower	9.5	32.7
Basin	14.5	49.9
Shower	24.5	76.9
L/Sink	13	44.7
K/Sink	13	44.7
Basin	44.5	109.2
Washing M/c	238.5	149.5
K/Sink	358	149.5
K/Sink	13	44.7
Basin	104.5	142.9
Bath	582	149.5
		Total q_{D_d} = 994.2

The heat loss during the periods of water flow could be readily calculated since it was reasonable to assume the forced-convection heat transfer coefficient would be 16 $\frac{W}{m^2 K}$.

$$q_{D_f} = h\ A_p\ (t_w - \bar{t}_a)\ T$$

$$= 16 \times 0.1005 \times (65.1-20.5) \times 25 \times 60 \times 10^{-3}$$

$$= 107.6\ \frac{kJ}{m}$$

Hence the daily total heat loss, with forced convection cooling was

$$q_D = 994.2 + 107.6$$

$$= 1101.8\ \frac{kJ}{m}$$

(B) Remaining pipe sections

Using the same procedures as in (A), the following results were obtained.

TABLE 5.5

Pipe Section	Daily Heat Loss in Forced Convection		
	Deadleg Losses	While Water was Flowing	Total
	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_{D_f} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$
J1-J2	430	45.4	475.4
J2-J3	226.2	28.7	255
J3-J4	175.6	31.8	207.4
J4-J5	81.8	21.8	103.6

(C) Comparison between natural and forced convection cooling results

The daily heat losses are summarised in Table 5.6 below, and the percentage difference ε is also shown.

$$\varepsilon \text{ is defined as } \varepsilon = \frac{\text{calculated-experimental}}{\text{calculated}} \times 100\%$$

TABLE 5.6

Pipe Section	Natural Convection Heat Loss	Forced Convection Heat Loss	% Difference ε
	[kJ/m per day]	[kJ/m per day]	
0-J1	1096.7	1101.8	-0.5
J1-J2	487.3	475.4	2.4
J2-J3	283.8	255	10
J3-J4	210.8	207.4	1.6
J4-J5	105.8	103.6	2

The % differences are seen to be within acceptable limits, showing that the air flow rates of the cooling air were set at the required values to give simulation.

5.5 Calculation of Actual Heat Loss With Forced Convection Cooling; Tank Temperature = 50°C Nominal

(A) All the sections of pipework

The calculations of Chapter 5.4 were repeated for a tank temperature of 50°C. The results are summarised below.

TABLE 5.7

Pipe Section	Daily Heat Loss in Forced Convection		
	Deadleg Losses	While Water Was Flowing	Total
	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_{D_f} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$
0-J1	588.6	64.8	653.4
J1-J2	253.2	25.5	278.7
J2-J3	147.8	18.2	166
J3-J4	105.3	19.1	124.4
J4-J5	52.9	14.5	67.4

(B) Comparison between natural and forced convection cooling results

TABLE 5.8

Pipe Section	Natural Convection Heat Loss	Forced Convection Heat Loss	% Difference ε
	[kJ/m per day]	[kJ/m per day]	
0-J1	679.4	653.4	3.8
J1-J2	299.2	278.7	6.9
J2-J3	176.4	166	5.9
J3-J4	131	124.4	5
J4-J5	63.3	67.4	-6.5

Again the % differences are within acceptable limits, indicating that the air flow settings were satisfactory.

5.6 Experimental Procedures and Test Conditions

Having established the necessary air flow rates, the experimental procedures were as below:

A. Tank temperature at $70^{o}C$ nominal
   1. Average water usage pattern with simulated deadleg cooling.
   2. High water usage pattern with simulated deadleg cooling.

B. Tank temperature at $50^{o}C$ nominal
   1. Average water usage pattern with simulated deadleg cooling
   2. High water usage pattern with simulated deadleg cooling.

C. Water quantity drawn off in different household devices
   1. With tank temperature $70^{o}C$
   2. With tank temperature $50^{o}C$.

Electricity input was recorded by a digital channel of the data logger in terms of pulses from a light sensor on the kWh meter. Hot water energy use was calculated by hot water quantity and temperature at the hot water outlet of each device. Tank loss was computed by the difference between the tank inlet and outlet temperatures, and hence the system loss was calculated by difference:

i.e. System Loss = Electrical Input - H.W. use - Tank Loss.

5.7 Comparison between natural and forced convection cooling results Under High Water Usage Pattern

The calculation techniques in the average water usage pattern were applied to the results with a high water usage pattern. The detailed calculations are presented in Appendices D and E. The comparison between natural and forced-convection cooling results under the high water usage pattern with various tank temperatures are listed in Tables 5.9 and 5.10.

(A) Tank Temperature = 70°C Nominal

TABLE 5.9

Pipe Section	Natural Convection Heat Loss	Forced Convection Heat Loss	% Difference ε
	[kJ/m per day]	[kJ/m per day]	
0-J1	1158.7	1167.5	-0.8
J1-J2	525	513.1	2.3
J2-J3	355.4	321.1	9.6
J3-J4	281.9	280.1	0.6
J4-J5	105.8	103.6	2

(B) Tank Tamperature = 50°C Nominal

TABLE 5.10

Pipe Section	Natural Convection Heat Loss	Forced Convection Heat Loss	% Difference ε
	[kJ/m per day]	[kJ/m per day]	
0-J1	716.1	688.4	3.9
J1-J2	321.4	300.6	6.5
J2-J3	220.9	207.9	5.9
J3-J4	175.3	169.2	3.5
J4-J5	63.3	67.4	-6.5

The % differences are seen to be within acceptable limits, showing that the same air flow rate at each pipe section can be used at both the high and average usage patterns with minimal error, but must be altered for a large change in water tank temperature (from 50°C to 70°C).

CHAPTER 6

6. COMPARISON AMONG CALCULATED, EXPERIMENTAL AND COMPUTER DATA

6.1 Comparison Between Calculated and Experimental Data

A summary of the calculated and experimental daily deadleg losses and total heat loss per metre pipe length (see Appendices D, E and Chapter 5) is listed in Table 6.1 (for 70°C tank temperature air flow setting) and in Table 6.3 (for 50°C tank temperature air flow setting). By multiplying the deadleg length of that section by the corresponding calculated and experimental daily heat losses per metre (see Table 6.1, 6.3), the daily deadleg losses $[Q_{D_d}]$ and total heat losses $[Q_D]$ were obtained and are listed in Table 6.2 (for high tank temperature) and Table 6.4 (for low tank temperature).

(A) Percentage difference between calculated and experimental values for high tank temperature (70°C) air flow setting (see Table 6.2)

(a) Average water usage pattern

(i) For system (O-J5) daily deadleg losses

$$\varepsilon = \frac{3227.3 - 3155.8}{3227.3} \times 100\% = 2.2\%$$

(ii) For system (O-J5) daily total heat losses

$$\varepsilon = \frac{3621.9 - 3555}{3621.9} \times 100\% = 1.8\%$$

(b) High water usage pattern

(i) For system (O-J5) daily deadleg losses

$$\varepsilon = \frac{3530.4 - 3460}{3530.4} \times 100\% = 2\%$$

(ii) For system (O-J5) daily total heat losses

$$\varepsilon = \frac{3976.1 - 3909.2}{3976.1} \times 100\% = 1.7\%$$

(B) Percentage difference between calculated and experimental values for low tank temperature ($50^{o}C$) air flow setting (see Table 6.4)

(a) Average water usage pattern

(i) For system (0-J5) daily deadleg losses

$$\varepsilon = \frac{1998.3 - 1899.2}{1998.3} \times 100\% = 5\%$$

(ii) For system (0-J5) daily total heat losses

$$\varepsilon = \frac{2232.6 - 2141}{2232.6} \times 100\% = 4.1\%$$

(b) High water usage pattern

(i) For system (0-J5) daily deadleg losses

$$\varepsilon = \frac{2182.8 - 2078.2}{2182.8} \times 100\% = 4.8\%$$

(ii) For system (0-J5) daily total heat losses

$$\varepsilon = \frac{2447.6 - 2349.8}{2447.6} \times 100\% = 4\%$$

TABLE 6.1 Heat Losses Per Metre Pipe Length: Calculated and Experimental, Daily Deadleg and Total Heat Losses for Initial Water Temperature at TC1 = 70°C, Under Various Water Usage Patterns

Section	Average Usage Pattern				High Usage Pattern			
	Calculated Loss		Experimental Loss		Calculated Loss		Experimental Loss	
	Deadleg	Total	Deadleg	Total	Deadleg	Total	Deadleg	Total
	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$
0-J1	983.4	1096.7	994.2	1101.8	1029.6	1158.7	1044.9	1167.5
J1-J2	440.3	487.3	430	475.4	472	525	461.9	513.1
J2-J3	252.5	283.8	226.2	255	318	355.4	286.8	321.1
J3-J4	184.3	210.8	175.6	207.4	249.8	281.9	241.4	280.1
J4-J5	88.5	105.8	81.8	103.6	88.5	105.8	81.8	103.6

TABLE 6.2 The System (0-J5) Calculated and Experimental Daily Deadleg and Total Heat Losses for Initial Water Temperature at TC1 = 70^{o}C, under Various Water Usage Patterns

Section	Average Usage Pattern				High Usage Pattern			
	Calculated Loss		Experimental Loss		Calculated Loss		Experimental Loss	
	Deadleg	Total	Deadleg	Total	Deadleg	Total	Deadleg	Total
	Q_{D_d} [kJ]	Q_D [kJ]	Q_{D_d} [kJ]	Q_D [kJ]	Q_{D_d} [kJ]	Q_D [kJ]	Q_{D_d} [kJ]	Q_D [kJ]
0-J1	1475.1	1645.1	1491.3	1652.7	1544.4	1738.1	1567.4	1751.3
J1-J2	924.6	1023.3	903	998.3	991.2	1102.5	970	1077.5
J2-J3	340.9	383.1	305.4	344.3	429.3	479.8	387.2	433.5
J3-J4	221.2	253	210.7	248.9	300	338.3	290	336.1
J4-J5	265.5	317.4	245.4	310.8	265.5	317.4	245.4	310.8
0-J5	3227.3	3621.9	3155.8	3555	3530.4	3976.1	3460	3909.2

TABLE 6.3 Heat Losses per Metre Pipe Length: Calculated and Experimental, Daily Deadleg and Total Heat Losses for Initial Water Temperature at TC1 = 50°C, Under Various Water Usage Patterns

Section	Average Usage Pattern				High Usage Pattern			
	Calculated Loss		Experimental Loss		Calculated Loss		Experimental Loss	
	Deadleg	Total	Deadleg	Total	Deadleg	Total	Deadleg	Total
	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$	q_{D_d} $\left[\frac{kJ}{m}\right]$	q_D $\left[\frac{kJ}{m}\right]$
0-J1	612.6	679.4	588.6	653.4	640	716.1	614.6	688.4
J1-J2	271	299.2	253.2	278.7	289.6	321.4	271.9	300.6
J2-J3	157.8	176.4	147.8	166	198.7	220.9	186.2	207.9
J3-J4	115	131	105.3	124.4	155.9	175.3	146	169.2
J4-J5	53.1	63.3	52.9	67.4	53.1	63.3	52.9	67.4

TABLE 6.4 The System (0-J5) Calculated and Experimental Daily Deadleg and Total Heat Losses for Initial Water Temperature at TC1 = 50°C, Under Various Water Usage Patterns

Section	Average Usage Pattern				High Usage Pattern			
	Calculated Loss		Experimental Loss		Calculated Loss		Experimental Loss	
	Deadleg	Total	Deadleg	Total	Deadleg	Total	Deadleg	Total
	Q_{D_d} [kJ]	Q_D [kJ]	Q_{D_d} [kJ]	Q_D [kJ]	Q_{D_d} [kJ]	Q_D [kJ]	Q_{D_d} [kJ]	Q_D [kJ]
0-J1	918.9	1019.1	822.9	980.1	960	1074.2	921.9	1032.6
J1-J2	569.1	628.3	531.7	585.3	608.2	674.9	571	631.3
J2-J3	213	238.1	199.5	224.1	268.2	298.2	251.4	280.7
J3-J4	138	157.2	126.4	149.3	187.1	210.4	175.2	203
J4-J5	159.3	189.9	158.7	202.2	159.3	189.9	158.7	202.2
0-J5	1998.3	2232.6	1899.2	2141	2182.8	2447.6	2078.2	2349.8

6.2 Comparison Between Experimental and Computer Data

(A) Predictions from computer result

TABLE 6.5 A Summary Result from the Computer Printouts in which 'System' Means 'TK-J5'

Printout	Mean Room Temperature	Hot Water		Electricity	System Eff.	System Loss
#	[°C]	[ℓ]	[kJ]	[kJ]	-	[kJ]
A1	23.4	180.1	34316	54526	0.629	5333.3
A2	21.2	215	41599	63535	0.655	6646.1
B1	20.9	222.7	27651	39187	0.706	3887
B2	23.1	259.2	31835	41570	0.766	4349.4

The printout number corresponds to the experimental routine as described in Chapter 5.6 in which experiment numbers A1, A2 are for water initial temperature at TC1 equal to 70°C and experiment numbers B1, B2 are for water initial temperature at TC1 equal to 50°C.

(B) Percentage water wasted for devices = Basin, L/K Sink, Bath

Data refer to Appendix F, Table F(2), F(3)

The % water wasted was calculated by

$$\%\ \text{water wasted} = \frac{\text{wasted water}}{\text{vol. of device}} \times 100\%$$

TABLE 6.6 Percentage of Water Wasted

Device	Vol. of Device	Wasted Water		% Water Wasted	
		70°C	50°C	70°C	50°C
	[ℓ]	[ℓ]	[ℓ]	[ℓ]	[ℓ]
Basin	3.9	0.76	1.2	19.5	30.8
L/K Sink	20.8	3.3	2.8	15.9	13.5
Bath	43.3	5.3	5.5	12.2	12.7

(C) <u>Comparison between experimental and computer analysis of total system heat losses</u>

System loss from computer results and experiment results are listed in Table 6.7 in which the computer results counted the heat loss for the whole hot-water pipe system while the experiment results are just concerned with the heat losses due to deadleg lengths in the system (0-J5). The percentage difference ε is defined as

$$\varepsilon = \frac{\text{computer-experimental}}{\text{computer}} \times 100\%$$

TABLE 6.7 <u>Comparison Between Computer (Whole) and Experimental (0-J5) System Heat Loss</u>

Experiment	System Loss		% Difference
	Computer $[Q_T]_{whole}$	Experimental $[Q_T]_{0-J5}$	ε
[#]	[kJ]	[kJ]	
A1	5333.3	3555	33
A2	6646.1	3909.2	41.2
B1	3887	2141	44.9
B2	4349.4	2349.8	46

CHAPTER 7

7. DISCUSSION AND CONCLUSION

Modifications have been made to the domestic hot water simulation rig which have added the feature of being able to simulate heat losses from the pipework. This has been achieved by placing 'tunnels' over the pipework through which cooled air was blown. Thus the natural convection cooling which normally occurs in dwellings has been simulated by forced convection cooling.

The cooled air was available at a temperature of approximately 10°C, and the flow rate to each tunnel was adjusted to give simulation. The simulation was established by comparing the calculated natural convection heat loss over a period of 24 hours when an average water pattern was used with the heat loss deduced from the experimental data. Tables 5.6 and 5.8 show these comparisons for tank water temperatures of 70°C (nominal) and 50°C (nominal) respectively. It can be seen that the differences are within acceptable limits, the largest being 10% for the pipe section J2-J3.

The rig was then operated with a different water usage pattern (a high usage pattern, see Appendix A), and the results have been tabulated in Chapter 5.7. The agreement between the losses expected from calculation and those deduced from the experimental data again is acceptable. The maximum % difference was 9.6 (see Appendices D, E) for pipe section J2-J3.

The higher heat loss for the computer calculated system as compared to the deadleg loss simulation system shows that a significant proportion of the total system heat loss (up to 46%) occurs from parts not included in the simulated deadleg system. It must be remember that the computed resuets were also subject to experimental error.

These parts are:

(a) The dropper from the hot water tank to the start of the deadleg simulation system. A temperature drop across this section from 1.5°C to 1.7°C was recorded.

(b) The risers running from the "deadleg simulation pipe" up to the various devices.

(c) Losses from fittings to the "deadleg simulation system".

7.1 Temperature Along the 'Horizontal' Deadleg Length

Three thermocouples were installed on each section of pipe with the objective of obtaining a reasonable average temperature, but it was found that, because of the insulating effect of the fittings at each end of the pipe sections, the end thermocouples frequently indicated temperatures slightly higher than the middle one. The middle thermocouple only was therefore used as a measure of the mean water temperature.

7.2 Cool Air Supply from the Refrigeration System

Due to the laboratory room temperature being higher than the air supply temperature to the refrigeration system in Summer, the dew point temperature was higher too. Therefore, ice started forming in the evaporator fins of the refrigerator during the middle of the 24-hour experimental runs and gradually, the temperature of the supply cool air became lower and lower. Although this problem could be fixed by adjusting the pressure of expansion valve, it was already in the upper limit of this heat pump system. As for the ice growing problem, a dehumidifier can be used to reduce the moisture from the supply air to the refrigeration unit.

7.3 The Proportion of System Deadleg Losses to Total Heat Losses

The system (0-J5) experimental deadleg and total heat losses for experiments A1, A2, B1 and B2 are listed below (see Table 6.2, 6.4). From these the percentage of deadleg losses to total system heat losses were calculated and are shown to be between 88% - 89%.

TABLE 7.1 The System (0-J5), Experimental Daily Deadleg and Total Heat Losses

Experiment	Deadleg Loss	System Heat Loss	$\frac{Q_{d_T}}{Q_T}$ x 100%
	Q_{d_T} [kJ]	Q_T [kJ]	[%]
A1	3155.8	3555	88.8
A2	3460	3909.2	88.5
B1	1899.2	2141	88.7
B2	2078.2	2349.8	88.4

7.4 Variation of Forced-Convection Heat Transfer Coefficient

From equation 5.1

$$\frac{-A_p h}{[TC]} T = \ln\left[\frac{t_w - t_a}{t_{w_o} - t_a}\right]$$

$$\exp. \frac{-A_p h}{[TC]} T = \frac{t_w - t_a}{t_{w_o} - t_a}$$

$$t_w - t_a = (t_{w_o} - t_a) \exp. \frac{-A_p h}{[TC]} T$$

$$\ln(t_w - t_a) = \left[\frac{-A_p h}{[TC]} \times (t_{w_o} - t_a)\right] \times T \tag{7.1}$$

There should be a linear relation between $\ln(t_w - t_a)$ and T, and the values of heat transfer coefficient should also be constant for forced-convection heat transfer as can be seen in equation 7.1. Shown in Table 7.2 are the experimental heat transfer coefficient for the experiment with tank temperature at $70^{o}C$ under various water usage patterns (see Chapter 5 and Appendix E). The maximum deviation from the average experimental heat transfer coefficient value is shown to be between 4% - 12.8%. Thus, the calculation based on forced convection equations in this project is applicable.

TABLE 7.2 Heat Transfer Coefficients of Cool Air for Tank Temperature = $70^{o}C$ Nominal

Time	Heat Transfer Coefficient				
T [min]	h $\left[\frac{W}{m^2k}\right]$				
	0-J1	J1-J2	J2-J3	J3-J4	J4-J5
0 - 20	17.6	10.6	9.2	12.6	14.4
20 - 40	16.3	9.9	9.7	12.3	13.6
40 - 60	16.5	9.8	10.2	11.8	13.6
60 - 80	15.7	9.6	10.1	12.4	13.3
80 - 100	16.2	9.6	9.7	11.7	13.1
100 - 120	14.8	8.8	10	11.3	13.4
120 - 140	14.9	8.7	9.4	-	-
140 - 160	15.7	9	-	-	-
160 - 180	-	9	-	-	-
180 - 200	-	8.7	-	-	-
$\bar{h}$	16	9.4	9.8	12	13.6
maximum deviation	10%	12.8%	4%	5.8%	5.9%

Photo 1 Air Conditioning Unit and PVC Air 'Tunnel'

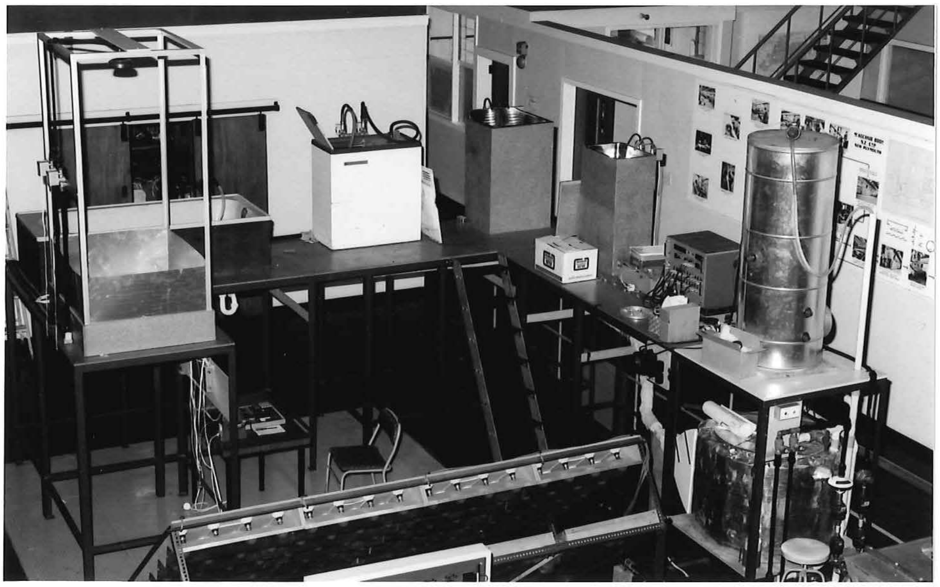

Photo 2 The Domestic Hot Water Dynamic Simulation Rig

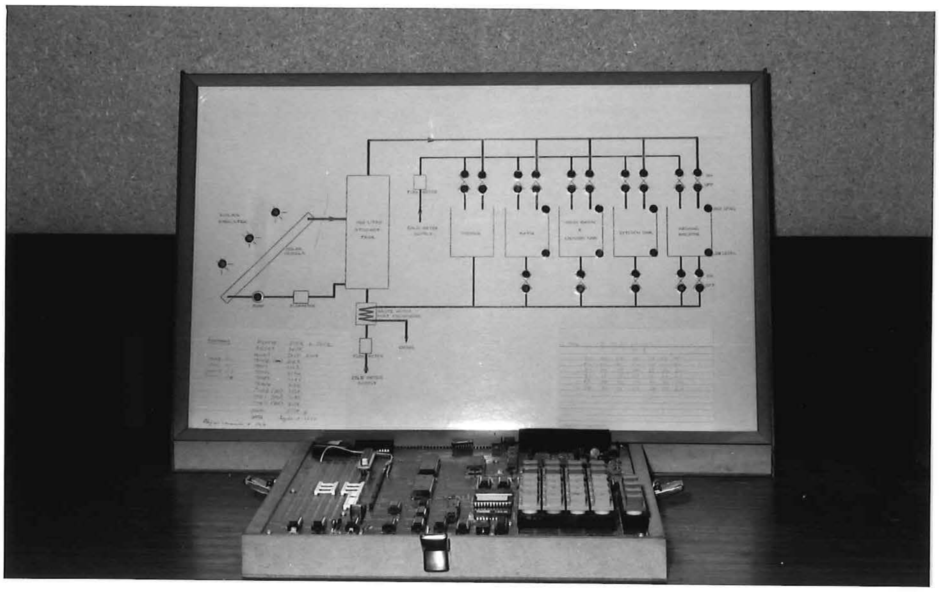

Photo 3 Z80 Starter Kit and Mimic Board

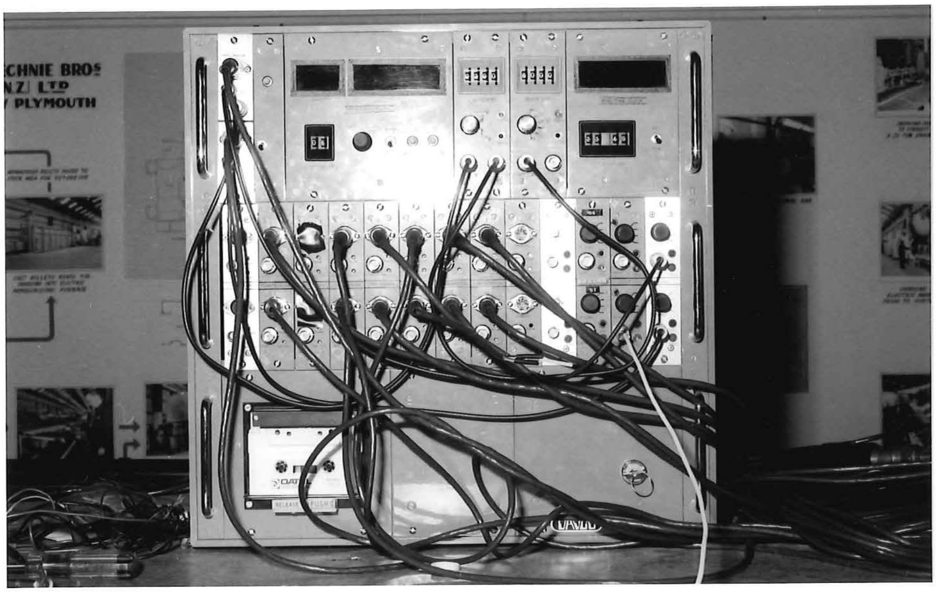

Photo 4 LPS-16 Data Logger

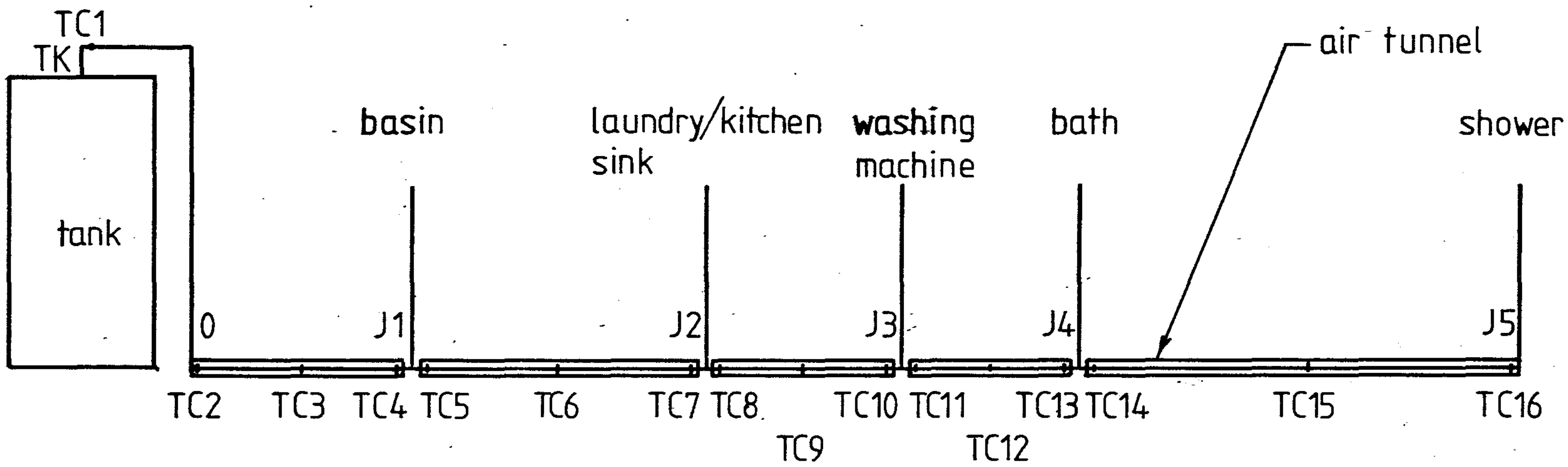

FIG 3·1 THE SCHEMATIC LAYOUT OF THE PIPEWORK ARRANGEMENT

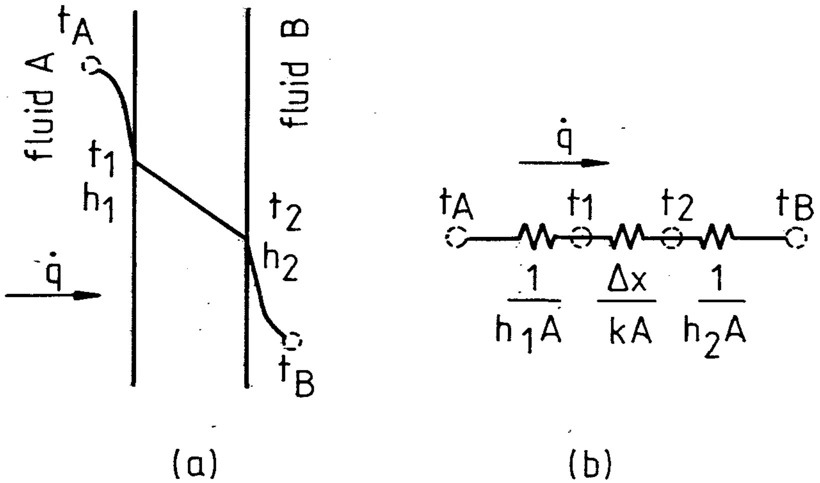

FIG 3·2 OVERALL HEAT TRANSFER THROUGH A PLANE WALL

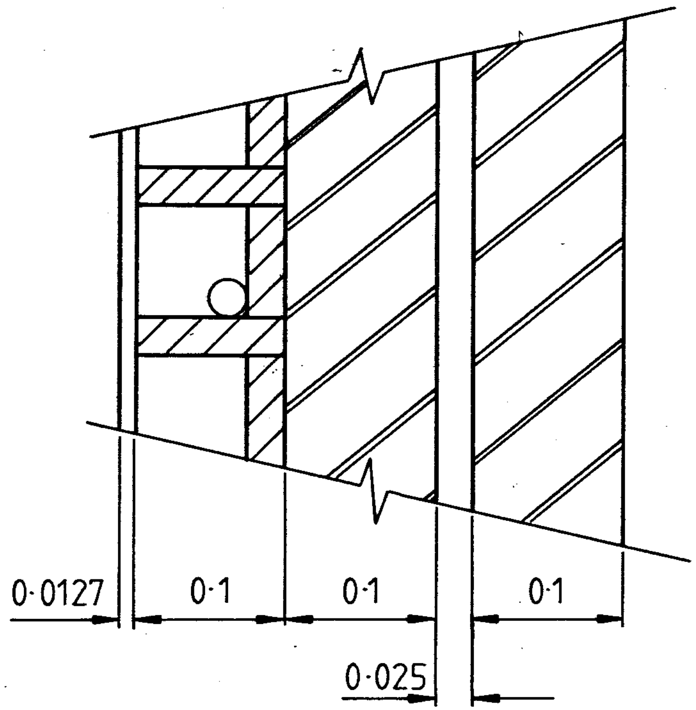

FIG 3·3 BUILDING FABRIC OF A DOMESTIC HOUSE

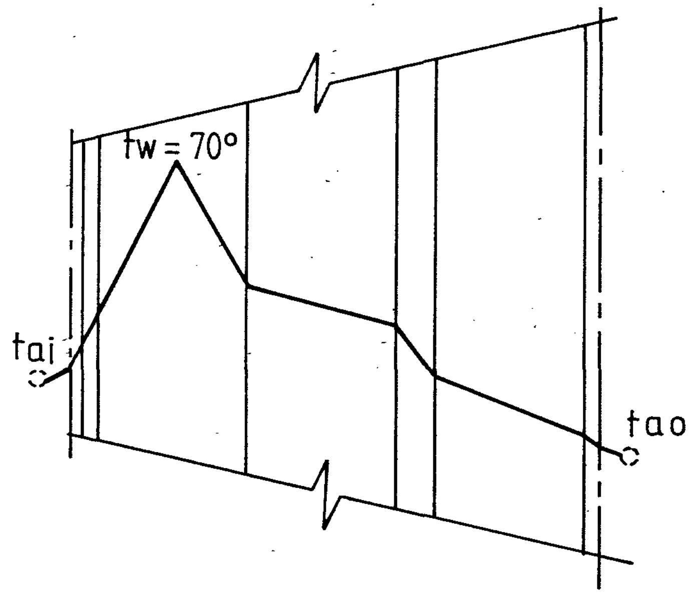

FIG 3·4(a) AIR-TO-AIR TEMPERATURE DIFFERENCE

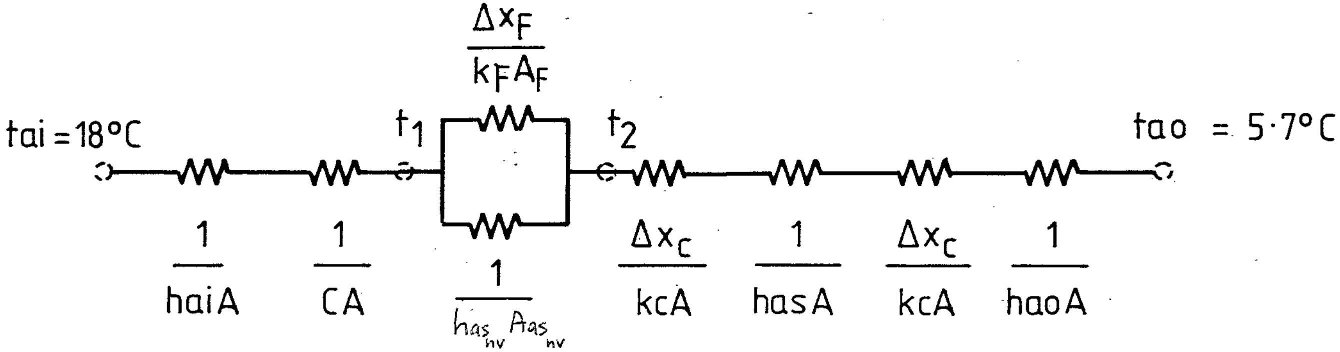

FIG 3·4(b) RESISTANCE ANALOGY FOR BUILDING FABRIC WITH CONVECTION BOUNDARIES

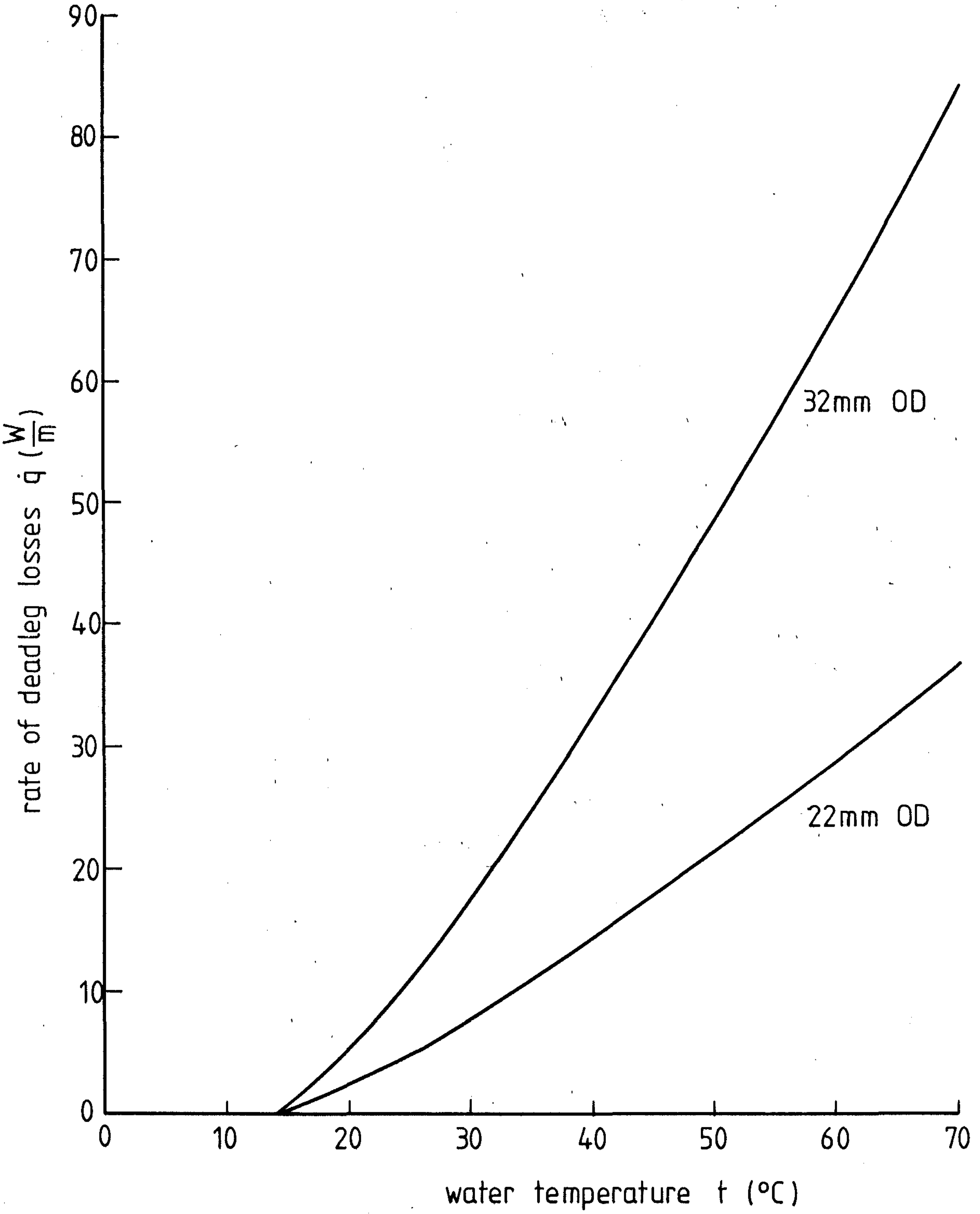

FIG 3·5 VARIOUS WATER DEADLEG LOSSES THROUGH BUILDING FABRIC.

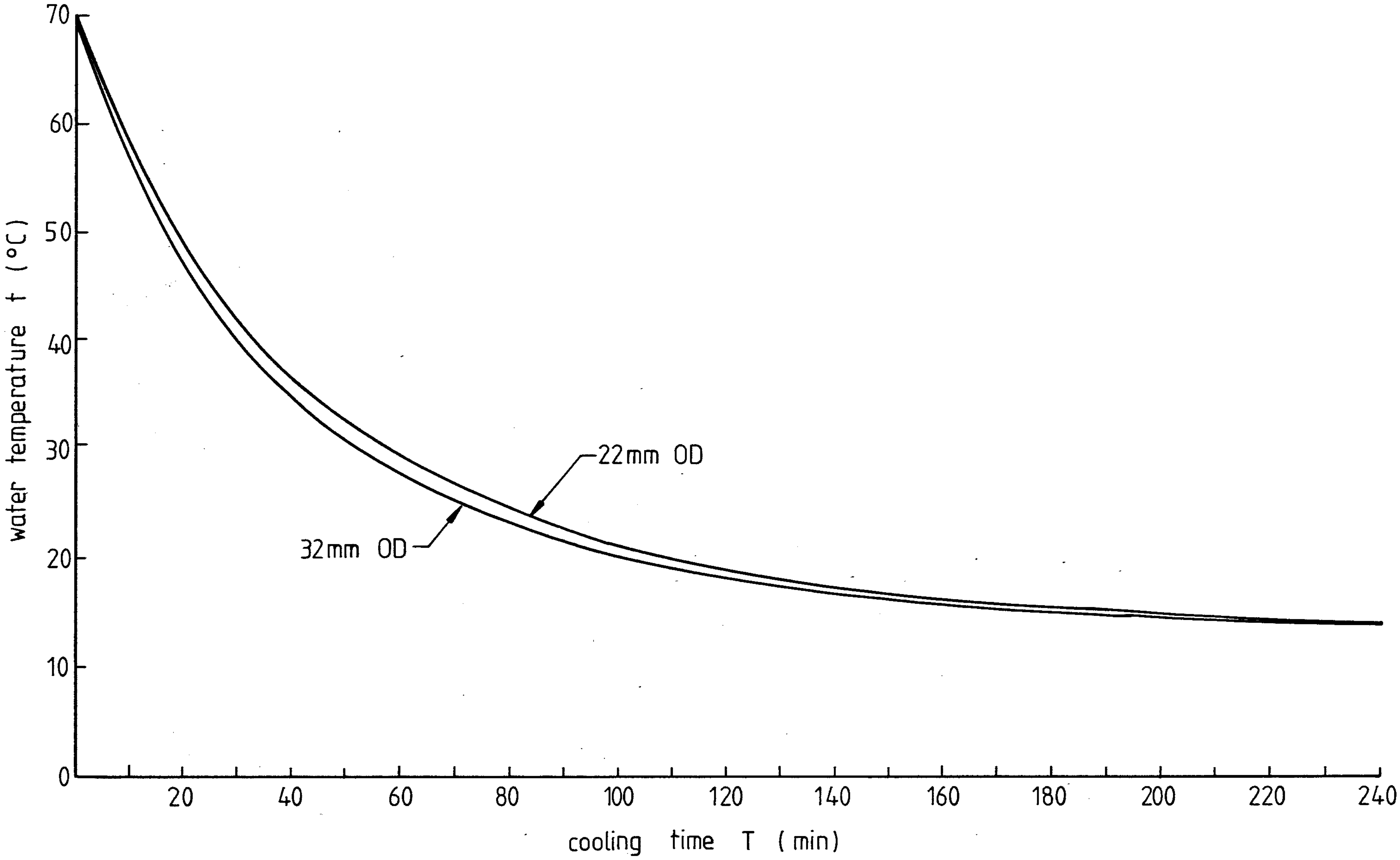

FIG 3·6 WATER COOLING RATE BY NATURAL CONVECTION WITH (i) OUTSIDE TEMPERATURE = 5·7°C
(ii) ROOM TEMPERATURE = 18°C

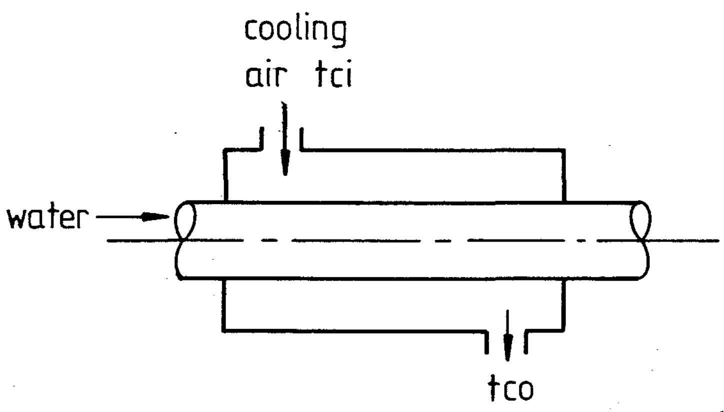

FIG 4·1 A SCHEMATIC LAYOUT OF THE SIMPLE PARALLEL FLOW HEAT EXCHANGER

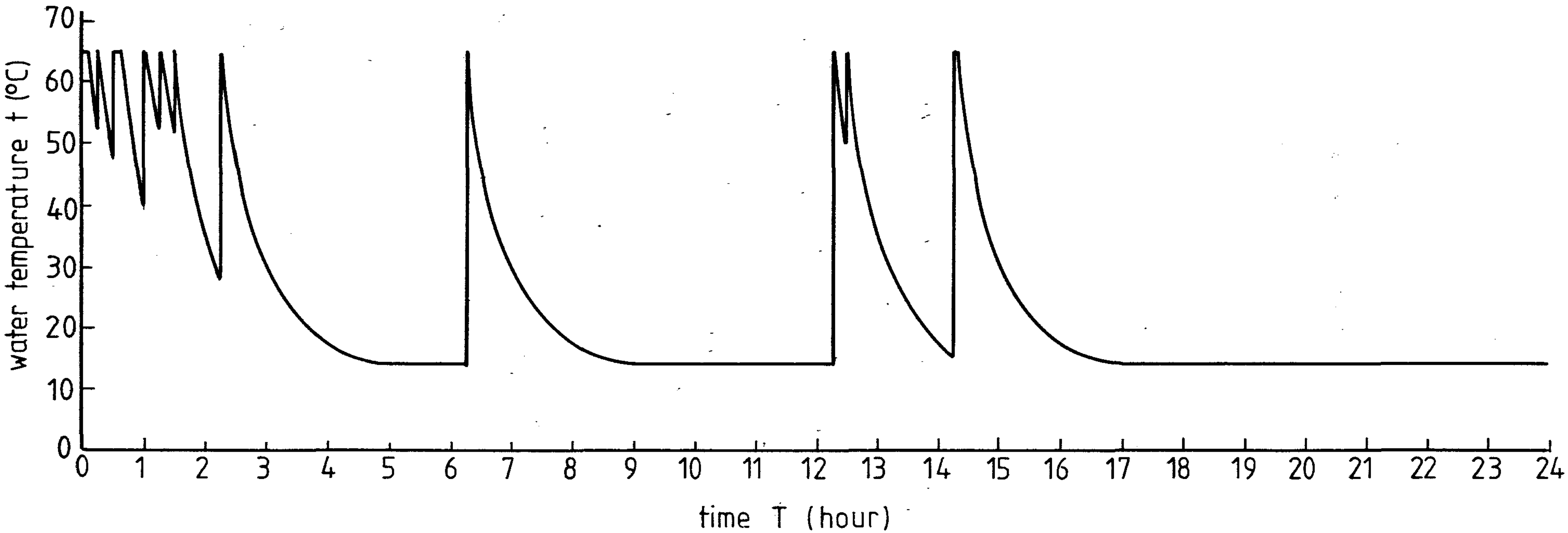

FIG 5·1 DAILY WATER TEMPERATURE PROFILE OF PIPE SECTION O–JI: (i) TANK TEMPERATURE = 70°C NOMINAL
(ii) AVERAGE WATER USAGE PATTERN

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APPENDIX A

HOT WATER FLOW IN THE FIVE PIPE SECTIONS UNDER VARIOUS WATER USAGE PATTERNS

According to the daily device usage sequence of hot water under average and high usage patterns from Reference [7] and the start-time of devices, the hot water flowing hold-time and cooling-time for the average water usage pattern of the five experimental pipe sections were calculated and are listed in Table A(1) to Table A(5), for the high water usage pattern are listed in Table A(6) to A(10).

A.1 Average Water Usage Pattern (AWUP)

TABLE A(1) AWUP with Hot Water Flow in Section 0-J1

Device	Start-Time	HW Flowing Hold-Time	HW Cooling Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	9.5
Basin	0.15	0.5	14.5
Shower	0.30	5.5	24.5
L/Sink	1.00	2	13
K/Sink	1.15	2	13
Basin	1.30	0.5	44.5
Washing M/c	2.15	1.5	238.5
K/Sink	6.15	2	358
K/Sink	12.15	2	13
Basin	12.30	0.5	104.5
Bath	14.15	3	582
		Total = 25	

TABLE A(2) AWUP With Hot Water Flow in Section J1-J2

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	24.5
L/Sink	1.00	2	13
K/Sink	1.15	2	58
Washing M/c	2.15	1.5	238.5
K/Sink	6.15	2	358
K/Sink	12.15	2	118
Bath	14.15	3	582
		Total = 23.5	

TABLE A(3) AWUP With Hot Water Flow in Section J2-J3

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	99.5
Washing M/c	2.15	1.5	718.5
Bath	14.15	3	582
		Total = 15.5	

TABLE A(4) AWUP With Hot Water Flow in Section J3-J4

Device	Start-Time	HW Flowing Hold-Time	HW Cooling Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	819.5
Bath	14.15	3	582
		Total = 14	

TABLE A(5) AWUP With Hot Water Flow in Section J4-J5

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	1410
		Total = 11	

A.2 High Water Usage Pattern (HWUP)

TABLE A(6) HWUP With Hot Water Flow in Section 0-J1

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	9.5
Basin	0.15	0.5	14.5
Shower	0.30	5.5	9.5
Basin	0.45	0.5	14.5
L/Sink	1.00	2	13
K/Sink	1.15	2	13
Basin	1.30	0.5	44.5
Washing M/c	2.15	1.5	238.5
K/Sink	6.15	2	358
K/Sink	12.15	2	13
Basin	12.30	0.5	14.5
Bath	12.45	3	87
Bath	14.15	3	582
		Total = 28.5	

TABLE A(7) HWUP With Hot Water Flow in Section J1-J2

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	24.5
L/Sink	1.00	2	13
K/Sink	1.15	2	58
Washing M/c	2.15	1.5	238.5
K/Sink	6.15	2	358
K/Sink	12.15	2	28
Bath	12.45	3	87
Bath	14.15	3	582
		26.5	

TABLE A(8) HWUP With Hot Water Flow in Section J2-J3

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	99.5
Washing M/c	2.15	1.5	628.5
Bath	12.45	3	87
Bath	14.15	3	582
		18.5	

TABLE A(9) HWUP With Hot Water Flow in Section J3-J4

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	729.5
Bath	12.45	3	87
Bath	14.15	3	582
		17	

TABLE A(10) HWUP With Hot Water Flow in Section J4-J5

Device	Start-Time	HW Flowing Hold-Time	HW Cooling-Time
	[Hr]	[Min]	[Min]
Shower	0.00	5.5	24.5
Shower	0.30	5.5	1410
		11	

Note: Data in Table A(10) is the same as in Table A(5).

APPENDIX B

TOTAL THERMAL CAPACITY OF HOT WATER AND PIPE

Thermal Capacity is

$$TC = C_{p_w} V_w \rho_w + C_{p_c} M_c$$

where the specific heat of water is 4180 $\frac{J}{kgK}$, the specific heat of copper is 392.5 $\frac{J}{kgK}$ and the water densities at various temperatures are as listed in Table B(1).

TABLE B(1) Water Densities with Various Temperatures

Water Temperature t [^oC]	Water Density ρ_w [kg/m^3]
70	977.5
60	983.3
50	988.1
40	992.2
30	995.6
20	998.2
10	999.7

For 0.030 m I.D. copper pipe with length L m

$$V_w = \frac{\pi}{4} (0.030)^2 \times L$$

$$= 7.07 \times 10^{-4} \, L \, m^3$$

$$M_c = 0.91 \text{ kg/m}$$

For 0.020 m I.D. copper pipe with length L m

$$V_w = \frac{\pi}{4} (0.020)^2 \times L$$

$$= 3.14 \times 10^{-4} \, L \, m^3$$

$$M_c = 0.71 \text{ kg/m}$$

Then, thermal capacities with various pipe sizes and water temperatures were calculated and are listed in Table B(2).

TABLE B(2) Thermal Capacities With Various Pipe Sizes and Water Temperatures

Water Temperature	Thermal Capacity	
	0.030 m I.D.	0.020 m I.D.
[°C]	[J/K]	[J/K]
70	3246 L	1560 L
60	3263 L	1570 L
50	3277 L	1576 L
40	3289 L	1580 L
30	3299 L	1586 L
20	3307 L	1589 L
10	3311 L	1591 L

Note: The values of thermal capacity, 3300 L J/K, for 0.030 m I.D. copper pipe and 1580 L J/K for 0.020 m I.D. copper pipe have been chosen for various water temperatures in the calculations of this report.

APPENDIX C

EXPERIMENTAL DATA

TABLE C(1) Experimental Data of:

(i) Air flow rate setting for high water temperature

(ii) Initial temperature at thermocouple # 1 ($t_i = 70^\circ C$)

(iii) Average air supplied temperature $\bar{t}_{a_s} = 12^\circ C$

(iv) Room Temperature = $20^\circ C$

Time T		SECTION 0-J1			SECTION J1-J2			SECTION J2-J3			SECTION J3-J4			SECTION J4-J5		
		Water Temp. t_w	Air Temp. $\bar{t}_a$	Temp. Diff. Δt	Water Temp. t_w	Air Temp. $\bar{t}_a$	Temp. Diff. Δt	Water Temp. t_w	Air Temp. $\bar{t}_a$	Temp. Diff. Δt	Water Temp. t_w	Air Temp. $\bar{t}_a$	Temp. Diff. Δt	Water Temp. t_w	Air Temp. $\bar{t}_a$	Temp. Diff. Δt
[min]	[sec]	[$^\circ$C]	[$^\circ$C]	[deg.C.]	[$^\circ$C]	[$^\circ$C]	[deg.C.]	[$^\circ$C]	[$^\circ$C]	[deg.C.]	[$^\circ$C]	[$^\circ$C]	[deg.C.]	[$^\circ$C]	[$^\circ$C]	[deg.C.]
0	0	65.1	20.5	44.6	65.5	21.5	44	66.2	20.5	45.7	63.6	20	43.6	56.8	23.5	33.3
10	600	51.4	19	32.4	53.9	20	33.9	54.1	19	35.1	49.6	17.5	32.1	42.7	20	22.7
30	1800	34.3	16	18.3	38.1	17.5	20.6	36.4	15.5	20.9	32	15	17	28.4	17	11.4
50	3000	25.4	15.5	9.9	28.9	16.5	12.4	26.7	15	11.7	23	13.5	9.5	21.1	15.5	5.6
70	4200	20.5	14.5	6	23.2	15.5	7.7	21	14	7	18.1	13.5	4.6	17.4	14.5	2.9
90	5400	17.6	14.5	3.1	19.6	15	4.6	17.5	13	4.5	15.3	12.5	2.8	15.5	14	1.5
110	6600	15.8	13.5	2.3	17.5	14	3.5	15.5	13	2.5	13.7	12	1.7	14.2	13.5	0.7
130	7800	14.8	13.5	1.3	16.3	14	2.3	14.3	12.5	1.8	-	-	-	-	-	-
150	9000	14.1	13.5	0.6	15.3	14	1.3	-	-	-	-	-	-	-	-	-
170	10200	-	-	-	14.8	14	0.8	-	-	-	-	-	-	-	-	-
190	11400	-	-	-	14.6	14	0.6	-	-	-	-	-	-	-	-	-

TABLE C(2) Experimental Data of:

(i) Air flow rate setting for low water temperature

(ii) Initial temperature at thermocouple # 1 ($t_i = 49.8^{\circ}C$)

(iii) Average air supplied temperature $\bar{t}_{a_s} = 14.6^{\circ}C$

(iv) Room Temperature = $24.5^{\circ}C$

Cooling Time T		SECTION 0-J1			SECTION J1-J2			SECTION J2-J3			SECTION J3-J4			SECTION J4-J5		
[min]	[sec]	Water Temp. t_w [°C]	Air Temp. $\bar{t}_a$ [°C]	Temp. Diff. Δt [deg.C.]	Water Temp. t_w [°C]	Air Temp. $\bar{t}_a$ [°C]	Temp. Diff. Δt [deg.C.]	Water Temp. t_w [°C]	Air Temp. $\bar{t}_a$ [°C]	Temp. Diff. Δt [deg.C.]	Water Temp. t_w [°C]	Air Temp. $\bar{t}_a$ [°C]	Temp. Diff. Δt [deg.C.]	Water Temp. t_w [°C]	Air Temp. $\bar{t}_a$ [°C]	Temp. Diff. Δt [deg.C.]
0	0	47.2	21	26.2	47.4	21.5	25.9	48	21.5	26.5	46.5	19.5	27	42.9	23	19.9
10	600	39.6	19.5	20.1	40.8	20	20.8	41.5	21	20.5	38.2	18.5	19.7	34.2	21	13.2
30	1800	29.8	18.5	11.3	31.2	18.5	12.7	31	18	13	27.4	16.5	10.9	25.9	19	6.9
50	3000	24.5	17.5	7	25.4	17.5	7.9	25	16.5	8.5	21.5	15.5	6	21.4	17.5	3.9
70	4200	21.4	17.5	3.9	22	17	5	21.5	16.5	5	18.4	15.5	2.9	19.2	17.5	1.7
90	5400	19.5	17.5	2	19.9	17	2.9	19.3	16	3.3	16.7	15	1.7	18.1	17	1.1
110	6600	18.4	17.5	0.9	18.6	17	1.6	17.9	15.5	2.4	15.7	15	0.7	17.5	17	0.5
130	7800	17.5	17	0.5	17.6	16.5	1.1	16.7	15.5	1.2	15	15	0	16.9	16.5	0.4
150	9000	17.1	17	0.1	17	16.5	0.5	16	15	1	14.6	14.5	0.1	16.6	16.5	0.1
170	10200	16.8	16.5	0.3	16.7	16.5	0.2	15.7	15.5	0.2	-	-	-	16.6	16.5	0.1

APPENDIX D

THE CALCULATED DAILY DEADLEG AND TOTAL HEAT LOSSES

D.1 Calculated Heat Losses for Initial Water Temperature, $t_i = 70^{\circ}C$ at Thermocouple # 1 Under Various Water Usage Patterns

(experimental data from Table C(1))

(A) Pipe section 0-J1 with $t_i = 65.1^{\circ}C$

(experimental data from thermocouple # 3)

(i) Deadleg Losses

TABLE D(1a)

Cooling-Time	Water Temperature	Rate of Heat Losses	Deadleg Losses
T [min]	t_w [$^{\circ}C$]	$\dot{q}$ [$\frac{W}{m}$]	q_d [$\frac{kJ}{m}$]
10	53	54.5	65.4
30	38	30	36
50	29.5	17	20.4
70	24	10	12
90	21	6.5	7.8
110	18.5	3.7	4.4
130	17	2.5	3.0
150	16	1.7	2
170	14.6	0.5	0.6
190	14.1	0.1	0.12
210	14.05	0.05	0.06
			$q_{d_T} = 151.8$

TABLE D(1b)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
9.5	31	9.5	31
14.5	47.4	14.5	47.4
24.5	73.5	9.5	31
13	42.5	14.5	47.4
13	42.5	13	42.5
44.5	106	13	42.5
238.5	151.8	44.5	106
358	151.8	238.5	151.8
13	42.5	358	151.8
104.5	142.6	13	42.5
582	151.8	14.5	47.4
	q_{D_d} = 983.4	87	136.5
		582	151.8
			q_{D_d} = 1029.6

(ii) Flowing Water Heat Losses

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 75.5 \frac{W}{m} \times 25 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$
$$= 113.3 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (983.4 + 113.3) \frac{kJ}{m}$$

$$= 1096.7 \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 75.5\ \frac{W}{m} \times 28.5\ \text{min} \times 60\ \frac{\text{sec}}{\text{min}} \times 10^{-3}\ \frac{kJ}{J}$$

$$= 129.1\ \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (1029.6 + 129.1)\ \frac{kJ}{m}$$

$$= 1158.7\ \frac{kJ}{m}$$

(B) <u>Pipe section J1-J2 with $t_i = 65.5^{\circ}C$</u>
(experimental data from thermocouple # 6)

(i) <u>Deadleg Losses</u>

TABLE D(2a)

Cooling-Time	Water Temperature	Rate of Heat Losses	Deadleg Losses
T [min]	t_w [$^{\circ}C$]	$\dot{q}$ [$\frac{W}{m}$]	q_d [$\frac{kJ}{m}$]
10	55	25	30
30	40	14.5	17.4
50	31	8.5	10.2
70	25.5	5.3	6.4
90	22	3.5	4.2
110	19.8	2.4	2.9
130	18	1.7	2
150	16.8	1.1	1.3
170	15.2	0.4	0.5
190	14.7	0.2	0.2
210	14.5	0.1	0.1
		Total q_{d_T}	= 75.2

TABLE D(2b)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	33.9	24.5	33.9
24.5	33.9	24.5	33.9
13	19.5	13	19.5
58	56.6	58	56.6
238.5	75.2	238.5	75.2
358	75.2	358	75.2
118	70.8	28	37
582	75.2	87	65.5
		582	75.2
	$q_{D_d} = 440.3$		$q_{D_d} = 472$

(ii) Flowing Water Heat Losses

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 33.3\ \frac{W}{m} \times 23.5 \text{ min} \times 60\ \frac{\text{sec}}{\text{min}} \times 10^{-3}\ \frac{kJ}{J}$$
$$= 47\ \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (440.3 + 47)\ \frac{kJ}{m}$$

$$= 487.3\ \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 33.3\ \frac{W}{m} \times 26.5 \text{ min} \times 60\ \frac{\text{sec}}{\text{min}} \times 10^{-3}\ \frac{kJ}{J}$$

$$= 53\ \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (472 + 53)\ \frac{kJ}{m}$$

$$= 525\ \frac{kJ}{m}$$

(C) Pipe section J2-J3 with $t_i = 66.2^{o}C$
(experimental data from thermocouple # 9)

(i) Deadleg Losses

The initial water temperature was nearly $65.5^{o}C$ so the data in Table 3.2a is applicable for this section.

TABLE D(3)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	33.9	24.5	33.9
99.5	68.2	99.5	68.2
718.5	75.2	628.5	75.2
582	75.2	87	65.5
	$q_{D_d} = 252.5$	582	75.2
			$q_{D_d} = 318$

(ii) Flowing Water Heat Losses

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 33.7 \frac{W}{m} \times 15.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 31.3 \frac{KJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (252.5 + 31.3) \frac{kJ}{m}$$

$$= 283.8 \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 33.7 \frac{W}{m} \times 18.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 37.4 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (318 + 37.4)\ \frac{kJ}{m}$$

$$= 355.4\ \frac{kJ}{m}$$

(D) Pipe section J3-J4 with $t_i = 63.6^{o}C$
(experimental data from thermocouple # 12)

(i) Deadleg Losses

Again, the initial water temperature was nearly $65.5^{o}C$, so the data in Table 3.2a is applicable for this section.

TABLE D.(4)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	33.9	24.5	33.9
819.5	75.2	729.5	75.2
582	75.2	87	65.5
		582	75.2
q_{D_d} = 184.3		q_{D_d} = 249.8	

(ii) Flowing Water Heat Losses

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 31.5\ \frac{W}{m} \times 14\ \text{min} \times 60\ \frac{sec}{min} \times 10^{-3}\ \frac{kJ}{J}$$

$$= 26.5\ \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (184.3 + 26.5)\ \frac{kJ}{m}$$

$$= 210.8\ \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 31.5 \frac{W}{m} \times 17 \text{ min} \times 60 \frac{\text{sec}}{\text{min}} \times 10^{-3} \frac{kJ}{J}$$
$$= 32.1 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (249.8 + 32.1) \frac{kJ}{m}$$

$$= 281.9 \frac{kJ}{m}$$

(E) Pipe section J4-J5 with $t_i = 56.8^{\circ}C$

(experimental data from thermocouple # 15)

(i) Deadleg Losses

TABLE D(5a)

Cooling-Time	Water Temperature	Rate of Heat Losses	Deadleg Losses
T [min]	t_w [$^{\circ}C$]	$\dot{q}$ [$\frac{W}{m}$]	q_d [$\frac{kJ}{m}$]
10	48.5	20	24
30	36	11.5	13.8
50	29	7.3	8.8
70	24	4.5	5.4
90	21	3	3.6
110	19	2	2.4
130	17.5	1.5	1.8
150	16.1	0.7	0.8
170	15	0.4	0.5
190	14.5	0.2	0.2
210	14.4	0.1	0.1
			$q_{d_T} = 61.4$

TABLE D(5b)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	27.1	24.5	27.1
1410	61.4	1410	61.4
$q_{D_d} = 88.5$		$q_{D_d} = 88.5$	

(ii) Flowing Water Heat Losses

For both water usage patterns, the total flowing water heat losses:

$$q_{D_f} = 26.2 \frac{W}{m} \times 11 \text{ min} \times 60 \frac{\text{sec}}{\text{min}} \times 10^{-3} \frac{kJ}{J}$$

$$= 17.3 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (88.5 + 17.3) \frac{kJ}{m}$$

$$= 105.8 \frac{kJ}{m}$$

D.2 Calculated Heat Losses for Initial Water Temperature $t_i = 49.8^\circ C$ at Thermocouple # 1 Under High Water Usage Pattern

(experimental data from Table C(2))

The calculated heat losses for initial water temperature $t_i = 50^\circ C$ at thermocouple # 1 under average water usage pattern are as listed in Chapter 5.3. In this section the heat losses under high water usage pattern are tabulated below.

(A) Pipe section 0-J1 with $t_i = 47.2°C$
(experimental data from thermocouple # 3)

(i) Deadleg Losses

TABLE D(6)

HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$
9.5	18.8
14.5	28.7
9.5	18.8
14.5	28.7
13	25.7
13	25.7
44.5	64.8
238.5	96.3
358	96.3
13	25.7
14.5	28.7
87	85.4
582	96.3
	$q_{D_d} = 640$

(ii) Flowing Water Heat Losses

The total flowing water heat losses:

$$q_{D_f} = 44.5 \frac{W}{m} \times 28.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 76.1 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (640 + 76.1) \frac{kJ}{m}$$

$$= 716.1 \frac{kJ}{m}$$

(B) Pipe section J1-J2 with $t_i = 47.4^\circ C$

(experimental data from thermocouple # 6)

(i) Deadleg Losses

TABLE D(7)

HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	20.4
24.5	20.4
13	9
58	34.7
238.5	47.3
358	47.3
28	22.3
87	40.9
582	47.3
	$q_{D_d} = 289.6$

(ii) Flowing Water Heat Losses

The total flowing water heat losses

$$q_{D_f} = 20 \frac{W}{m} \times 26.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 31.8 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (289.6 + 31.8) \frac{kJ}{m}$$

$$= 321.4 \frac{kJ}{m}$$

(C) Pipe section J2-J3 with $t_i = 48^{o}C$
(experimental data from thermocouple # 9)

(i) Deadleg Losses

TABLE D(8)

HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	20.4
99.5	42.8
628.5	47.3
87	40.9
582	47.3
	$q_{D_d} = 198.7$

(ii) Flowing Water Heat Losses

The total flowing water heat losses:

$$q_{D_f} = 20 \frac{W}{m} \times 18.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 22.2 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (198.7 + 22.2) \frac{kJ}{m}$$

$$= 220.9 \frac{kJ}{m}$$

(D) Pipe section J3-J4 with $t_i = 46.5^{\circ}C$

(experimental data from thermocouple # 12)

(i) Deadleg Losses

TABLE D(9)

HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T [min]	q_d $[\frac{kJ}{m}]$
24.5	20.4
729.5	47.3
87	40.9
582	47.3
	$q_{D_d} = 155.9$

(ii) Flowing Water Heat Losses

The total flowing water heat losses:

$$q_{D_f} = 19 \frac{W}{m} \times 17 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 19.4 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (155.9 + 19.4) \frac{kJ}{m}$$

$$= 175.3 \frac{kJ}{m}$$

(E) Pipe section J4-J5 with $t_i = 42.9^{\circ}C$

(experimental data from thermocouple # 15)

The deadleg losses and total heat losses would be the same as for the average water usage pattern as listed in Chapter 5.3.

APPENDIX E

THE EXPERIMENTAL DAILY DEADLEG AND TOTAL HEAT LOSSES

E.1 Experimental Heat Losses for Initial Water Temperature, $t_i = 70^{\circ}C$ at Thermocouple # 1 Under Various Water Usage Patterns. (experimental data from Table C(1))

(A) Pipe Section 0-J1

$$TC = 3300\ L\ \frac{J}{K}$$

$$C = \ln 44.6 = 3.8$$

$$A_p = 0.1005\ L\ m^2$$

(i) Deadleg Losses

TABLE E(1a)

Cooling-Time	Heat-Transfer Coefficient	Deadleg Losses
T	h	q_d
[min]	$\left[\frac{W}{m^2 K}\right]$	$\left[\frac{kJ}{m}\right]$
10	17.6	68.8
30	16.3	36
50	16.5	19.7
70	15.7	11.4
90	16.2	6.1
110	14.8	4.1
130	14.9	2.3
150	15.7	1.1
		$q_{d_T} = 149.5$

TABLE E(1b)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
9.5	32.7	9.5	32.7
14.5	49.9	14.5	49.9
24.5	76.9	9.5	32.7
13	44.7	14.5	49.9
13	44.7	13	44.7
44.5	109.2	13	44.7
238.5	149.5	44.5	109.2
358	149.5	238.5	149.5
13	44.7	358	149.5
104.5	142.9	13	44.7
582	149.5	14.5	49.9
q_{D_d} = 994.2		87	138
		582	149.5
		q_{D_d} = 1044.9	

(ii) Flowing Water Heat Losses

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 16 \frac{w}{m^2K} \times 0.1005 \frac{m^2}{m} \times 44.6 \text{ deg. C.} \times 25 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 107.6 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (994.2 + 107.6) \frac{kJ}{m}$$

$$= 1101.8 \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 16 \frac{W}{m^2 K} \times 0.1005 \frac{m^2}{m} \times 44.6 \text{ deg. C.} \times 28.5 \text{ min} \times 60 \frac{\text{sec}}{\text{min}} \times 10^{-3} \frac{kJ}{J}$$

$$= 122.6 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (1044.9 + 122.6) \frac{kJ}{m}$$

$$= 1167.5 \frac{kJ}{m}$$

(B) Pipe section J1-J2

$TC = 1580 \; L \; \frac{J}{K}$

$C = \ln 44 = 3.8$

$A_p = 0.069 \; L \; m^2$

(i) Deadleg Losses

TABLE E(2a)

Cooling-Time	Heat-Transfer Coefficient	Deadleg Losses
T [min]	h $\left[\frac{W}{m^2 K}\right]$	q_d $\left[\frac{kJ}{m}\right]$
10	10.6	29.8
30	9.9	16.9
50	9.8	10.1
70	9.6	6.1
90	9.6	3.7
110	8.8	2.6
130	8.7	1.7
150	9	1
170	9	0.6
190	8.7	0.4
		$q_{d_T} = 72.9$

TABLE E(2b)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	33.6	24.5	33.6
24.5	33.6	24.5	33.6
13	19.4	13	19.4
58	55.8	58	55.8
238.5	72.9	238.5	72.9
358	72.9	358	72.9
118	68.9	28	36.6
582	72.9	87	64.2
		582	72.9
$q_{D_d} = 430$		$q_{D_d} = 461.9$	

(ii) <u>Flowing Water Heat Losses</u>

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 10.6 \frac{W}{m^2K} \times 0.069 \frac{m^2}{m} \times 44 \text{ deg. C.} \times 23.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 45.4 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (430 + 45.4) \frac{kJ}{m}$$
$$= 475.4 \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses

$$q_{D_f} = 10.6 \frac{W}{m^2K} \times 0.069 \frac{m^2}{m} \times 44 \text{ deg. C.} \times 26.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 51.2 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (461.9 + 51.2)\ \frac{kJ}{m}$$

$$= 513.1\ \frac{kJ}{m}$$

(C) Pipe section J2-J3

$$TC = 1580\ L\ \frac{J}{K}$$

$$C = \ln 45.7 = 3.8$$

$$A_p = 0.069\ L\ m^2$$

(i) Deadleg Losses

TABLE E(3a)

Cooling-Time	Heat-Transfer Coefficient	Deadleg Losses
T [min]	h $\left[\frac{W}{m^2K}\right]$	q_d $\left[\frac{kJ}{m}\right]$
10	9.2	26.7
30	9.7	16.8
50	10.2	9.9
70	10.1	5.9
90	9.7	3.6
110	10	2.1
130	9.4	1.4
		$q_{d_T} = 66.4$

TABLE E(3b)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	30.5	24.5	30.5
99.5	62.9	99.5	62.9
718.5	66.4	628.5	66.4
582	66.4	87	60.6
		582	66.4
	$q_{D_d} = 226.2$		$q_{D_d} = 286.8$

(ii) Flowing Water Heat Losses

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 9.8 \frac{W}{m^2 K} \times 0.069 \frac{m^2}{m} \times 45.7 \text{ deg. C.} \times 15.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 28.7 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (226.2 + 28.7) \frac{kJ}{m}$$

$$= 255 \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 9.8 \frac{W}{m^2 K} \times 0.069 \frac{m^2}{m} \times 45.7 \text{ deg. C.} \times 18.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 34.3 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (286.8 + 34.3) \frac{kJ}{m}$$

$$= 321.1 \frac{kJ}{m}$$

(D) Pipe section J3-J4

$TC = 1580\ L\ \frac{J}{K}$

$C = \ln 43.6 = 3$

$A_p = 0.069\ L\ m^2$

(i) Deadleg Losses

TABLE E(4a)

Cooling-Time	Heat-Transfer Coefficient	Deadleg Losses
T [min]	h $\left[\frac{W}{m^2 K}\right]$	q_d $\left[\frac{kJ}{m}\right]$
10	12.6	33.5
30	12.3	17.3
50	11.8	9.3
70	12.4	4.7
90	11.7	2.7
110	11.3	1.6
		$q_{d_T} = 69.1$

TABLE E(4b)

AVERAGE USAGE PATTERN		HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses	Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$	T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	37.4	24.5	37.4
819.5	69.1	729.5	69.1
582	69.1	87	65.8
		582	69.1
	$q_{D_d} = 175.6$		$q_{D_d} = 241.4$

(ii) Flowing Water Heat Losses

(a) For average water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 12.6 \frac{w}{m^2 K} \times 0.069 \frac{m^2}{m} \times 43.6 \text{ deg. C.} \times 14 \text{ min} \times 60 \frac{\text{sec}}{\text{min}} \times 10^{-3} \frac{kJ}{J}$$

$$= 31.8 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses

$$q_D = (175.6 + 31.8) \frac{kJ}{m}$$

$$= 207.4 \frac{kJ}{m}$$

(b) For high water usage pattern, the total flowing water heat losses:

$$q_{D_f} = 12.6 \frac{w}{m^2 K} \times 0.069 \frac{m^2}{m} \times 43.6 \text{ deg. C.} \times 17 \text{ min} \times 60 \frac{\text{sec}}{\text{min}} \times 10^{-3} \frac{kJ}{J}$$

$$= 38.7 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (241.4 + 38.7) \frac{kJ}{m}$$

$$= 280.1 \frac{kJ}{m}$$

(E) Pipe section J4-J5

$$TC = 1580 \text{ L} \frac{J}{K}$$

$$C = \ln 33.3 = 3.5$$

$$A_p = 0.069 \text{ L m}^2$$

(i) Deadleg Losses

TABLE E(5a)

Cooling-Time	Heat-Transfer Coefficient	Deadleg Losses
T	h	q_d
[min]	$\left[\frac{W}{m^2 K}\right]$	$\left[\frac{kJ}{m}\right]$
10	14.4	27.1
30	13.6	12.8
50	13.6	6.3
70	13.3	3.2
90	13.1	1.6
110	13.4	0.8
		q_{d_T} = 51.8

TABLE E(5b)

AVERAGE AND HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T	q_d
[min]	$\left[\frac{kJ}{m}\right]$
24.5	30
1410	51.8
	q_{D_d} = 81.8

(ii) Flowing Water Heat Losses

For both water usage pattern, the total flowing heat losses:

$$q_{D_f} = 14.4 \frac{W}{m^2 K} \times 0.069 \frac{m^2}{m} \times 33.3 \text{ deg. C.} \times 11 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 21.8 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (81.8 + 21.8) \frac{kJ}{m}$$

$$= 103.6 \frac{kJ}{m}$$

E.2 Experimental Heat Losses for Initial Water Temperature $t_i = 49.8^{\circ}C$ at Thermocouple # 1 Under High Water Usage Pattern (experimental data from Table C(2))

The experimental heat losses for initial water temperature $t_i = 49.8^{\circ}C$ at thermocouple # 1 under average water usage pattern are as listed in Chapter 5.5. The heat losses under high water usage pattern are tabulated here.

(A) Pipe section 0-J1

(i) Deadleg Losses

TABLE E(6)

HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$
9.5	18.9
14.5	28.9
9.5	18.9
14.5	28.9
13	25.9
13	25.9
44.5	64.4
238.5	88.5
358	88.5
13	25.9
14.5	28.9
87	82.5
582	88.5
	$q_{D_d} = 614.6$

(ii) Flowing Water Heat Losses

$$q_{D_f} = 16.4 \frac{W}{m^2 K} \times 0.1005 \frac{m^2}{m} \times 26.2 \text{ deg. C.} \times 28.5 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

$$= 73.8 \frac{kJ}{m}$$

Hence the total deadleg and flowing hot water heat losses:

$$q_D = (614.6 + 73.8) \frac{kJ}{m}$$

$$= 688.4 \frac{kJ}{m}$$

(B) Pipe section J1-J2

(i) Deadleg Losses

TABLE E(7)

HIGH USAGE PATTERN	
Cooling- Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	19.7
24.5	19.7
13	11.3
58	33.2
238.5	42.7
358	42.7
28	21.5
87	38.4
582	42.7
	q_{D_d} = 271.9

(ii) Flowing Water Heat Losses

$$q_{D_f} = 10.1 \frac{W}{m^2 K} \times 0.069 \frac{m^2}{m} \times 25.9 \text{ deg. C.} \times 26.5 \text{ min} \times 60 \frac{\text{sec}}{\text{min}} \times 10^{-3} \frac{kJ}{J}$$

$$= 28.7 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (271.9 + 28.7) \frac{kJ}{m}$$

$$= 300.6 \frac{kJ}{m}$$

(C) Pipe section J2-J3

(i) Deadleg Losses

TABLE E(8)

HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	20.4
99.5	40.2
628.5	43.6
87	38.4
582	43.6
	q_{D_d} = 186.2

(ii) Flowing Water Heat Losses

$$q_{D_f} = 10.7 \frac{W}{m^2 K} \times 0.069 \frac{m^2}{m} \times 26.5 \text{ deg. C} \times 18.5 \text{ min} \times 60 \frac{\text{sec}}{\text{min}} \times 10^{-3} \frac{kJ}{J}$$

$$= 21.7 \frac{kJ}{m}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (186.2 + 21.7) \frac{kJ}{m}$$

$$= 207.9 \frac{kJ}{m}$$

(D) Pipe section J3-J4

(i) Deadleg Losses

TABLE E(9)

HIGH USAGE PATTERN	
Cooling-Time	Deadleg Losses
T [min]	q_d $\left[\frac{kJ}{m}\right]$
24.5	20.4
729.5	43.6
87	38.4
582	43.6
	$q_{D_d} = 146$

(ii) Flowing Water Heat Losses

$$q_{D_f} = 12.2 \frac{W}{m^2 K} \times 0.069 \frac{m^2}{m} \times 27 \text{ deg. C.} \times 17 \text{ min} \times 60 \frac{sec}{min} \times 10^{-3} \frac{kJ}{J}$$

Hence, the total deadleg and flowing hot water heat losses:

$$q_D = (146 + 23.2) \frac{kJ}{m}$$

$$= 169.2 \frac{kJ}{m}$$

(E) For section J4-J5

The deadleg losses and total heat losses would be the same as for the average water usage pattern as noted in Chapter 5.5(A).

APPENDIX F

WASTED WATER FOR DEVICES: BASIN, L/K SINK, BATH

TABLE F(1) Water Consumption of Each Device from Computer Data

Device	70°C		50°C	
	H.W.	C.W.	H.W.	C.W.
	[ℓ]	[ℓ]	[ℓ]	[ℓ]
Basin	2.9	1.7	3.4	1.7
L/K Sink	15.5	8.6	18.5	5.1
Bath	29.2	19.4	38.4	10.4

TABLE F(2) Experimental Data

Device	Volume of Device
	[ℓ]
Basin	3.9
L/K Sink	20.8
Bath	43.3

Wasted Water

Data refer to Table F(1), F(2)

Wasted water for each device = (H.W. + C.W.) - vol. of device.

TABLE F(3) Wasted Water for Each Device

Device	Wasted Water	
	70°C	50°C
	[ℓ]	[ℓ]
Basin	0.76	1.2
L/K Sink	3.3	2.8
Bath	5.3	5.5